PETROL MOTORS AND MOTOR CARS

A HANDBOOK FOR
ENGINEERS, DESIGNERS, AND DRAUGHTSMEN

BY

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WITH ILLUSTRATIONS

SECOND EDITION

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PETROL MOTORS AND MOTOR CARS

The historical and theoretical aspects have only been lightly treated. Both have been ably dealt with by more competent hands than the present writer's, and, moreover, they are better kept apart from such a work as this.

The writer's thanks are due, and are heartily accorded, to Mr. E. J. Stoddard, of Detroit, U.S.A., for his permission to include some of his work. The writer is indebted to him for the formula for the design of cylinders (p. 12), and for his diagram (Fig. 6) of pressures and volumes. Probably many of the formulæ will be recognized as old friends, though, perhaps, in new disguises. As far as possible errors have been eliminated by careful checking, and it is thought that the book may be relied upon to give good working results. In this matter of revision material assistance has been rendered the writer by Mr. G. W. Sinclair, notably with regard to the mathematics.

It has not been thought necessary to give the derivation of the various formulæ, which would not affect their utility.

In addition to the general index, all the formulæ have been separately indexed to facilitate reference.

T. HYLER WHITE.

LONDON, 1904.

PREFACE TO THE SECOND EDITION.

THE cordial reception accorded the first edition of this book by the press and automobile public has given much pleasure to the writer, who takes this opportunity of expressing his thanks.

It seems to have been expected that examples of present-day designs should have been included for the guidance of designers. The writer ventures to think that, owing to the constant and rapid changes in automobile designs, such examples would quickly become out of date. Moreover, designers are already well served in this direction by the various journals devoted to automobile matters. Therefore, although several additions have been made to this second edition, giving data relating to the leading makes of petrol motors and speed gears, the original aim of the book has been adhered to, *i.e.* to enunciate principles rather than illustrate individual designs.

Another point raised is that the book does not cover the whole of the subject of the motor car of to-day. It is the writer's experience that the matters treated upon are those which are most often required by the motor designer, and it has been thought preferable to confine

the book to these points, and thereby keep it within handy limits.

The Table of Data relating to Typical Motors is inserted by the courtesy of Mr. Robert E. Phillips, and will show the extraordinary variation in present-day motor design.

T. HYLER WHITE.

LONDON, 1905.

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PART I.

PETROL MOTORS AND MOTOR CARS.

PETROL MOTORS.

INTRODUCTORY.

1. The Beau de Rochas, or Otto Cycle.—It was in 1862 that M. Beau de Rochas, a French engineer, patented an internal combustion engine, the principles of which have formed the basis for designers of this class of engine ever since. The conditions laid down in the patent, upon which the success of the engine depended, were—

(*a*) Maximum cylinder capacity, with a minimum of circumferential surface.
(*b*) High piston speed.
(*c*) Greatest possible compression.
(*d*) Maximum pressure at the commencement of the power stroke.

With the exception of *b*, these conditions have been embodied in all the most successful internal combustion engines. The exception, high piston speed, has also been adopted to a certain extent. It has been found that the piston speed is limited in practice.

The Beau de Rochas cycle is often miscalled the Otto cycle, chiefly because Dr. N. A. Otto was the first to make practical use of it in an engine, but the whole credit of the invention is certainly due to Beau de Rochas.

The series of operations in an engine working on this cycle are shown diagrammatically in Fig. 1. The indicator diagram, shown above the cylinder, will enable the action of the gases, in relation to the movements of the piston, to be followed. Similar letters are used to denote corresponding points in the cycle on the crank path, piston travel, and indicator diagram. As it requires two complete revolutions of the crank shaft for completion of the cycle, the crank path has been drawn as two circles, to prevent overlapping.

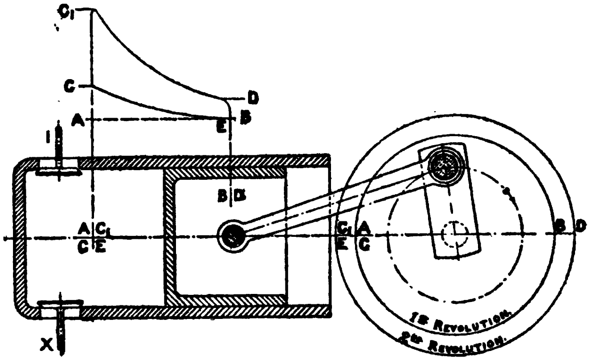


Fig. 1.

During the first out-stroke of the piston, from A to B, a charge of air and gas is drawn into the cylinder through the inlet valve I. From B to C, the first in-stroke, the charge is compressed. At about point C the compressed charge is ignited, and the consequent rise in pressure causes the piston to make its second out-stroke (power stroke) from C to D. During the second in-stroke, D to E, the exhaust valve X is held open, and the products of combustion are expelled, thus completing the cycle.

To avoid infringing the patent rights which covered this cycle, many attempts were made to construct engines

working on a different cycle. Of these, that patented by Mr. Dugald Clerk, in 1880, was one of the most successful. In this engine the suction and exhaust strokes of the Beau de Rochas cycle were eliminated. A separate pump was employed to force the charge of air and gas into the cylinder at the termination of the expansion stroke, sweeping the products of combustion out through ports in the cylinder wall, which were uncovered by the piston towards the end of each out-stroke. This charge was compressed by the in-stroke of the piston and ignited in the usual manner, and thus a power stroke was obtained once in every revolution of the crank shaft.

Two-stroke-cycle Engines.—The Beau de Rochas cycle requires four strokes of the piston to complete it, and a power stroke is only obtained once in every alternate revolution of the crank shaft. Not only because of the patent rights, but to obtain a more even turning moment, two-stroke-cycle engines have received a good deal of attention from time to time. The present type of two-stroke-cycle engines have been evolved from the Clerk engine, and the operation of a typical example is shown diagrammatically in Fig. 2. The reference letters are used in a similar sense to those in Fig. 1, two separate circles being used for the crank path, the inner one showing the operations taking place in the crank chamber, and the outer one those occurring in the cylinder. By the aid of the indicator diagrams shown above the cylinder and the crank chamber, the behaviour of the gases can be readily followed.

During a part of the first in-stroke of the piston, from A to B, a charge of air and gas is drawn into the crank chamber, through the check valve at V. During a portion of the first out-stroke, from B to C, the piston compresses the charge within the crank chamber to about five pounds per square inch above atmosphere, until the piston has

moved far enough to uncover the port I in the cylinder wall, when a considerable part of the compressed charge flows into the cylinder (C to A), through the transfer pipe shown. This charge is then further compressed by the next in-stroke of the piston, from D to E, and is ignited, as in a four-cycle engine, causing the piston to make a power out-stroke from F to G. Just before the inlet port I is uncovered by the piston, the exhaust port X is uncovered, thus allowing the products of combustion to

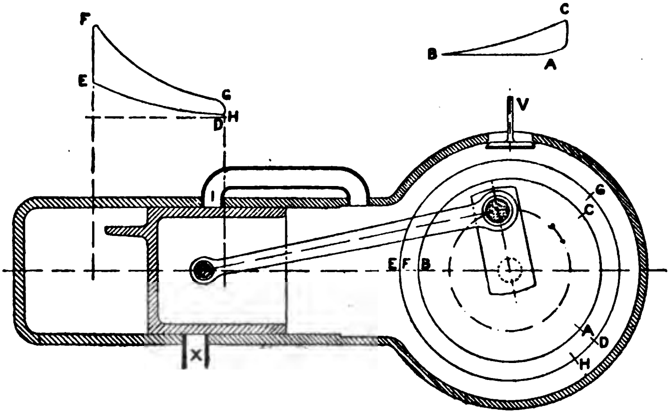


FIG. 2.

escape, G to H. By the time the pressure in the cylinder has dropped to somewhere about that of the atmosphere, the inlet port I is again uncovered, and a new charge enters the cylinder. The fresh charge is prevented from passing across the piston, and so escaping through the exhaust port X, by the baffle plate, cast on the piston head, which deflects the incoming charge towards the top of the cylinder, and thus assists in sweeping the products of combustion out through the exhaust port. The operations in the crank chamber are simultaneous with those in the

cylinder, as will be evident from inspection of the two crank-path circles. Thus while the fresh charge is being drawn into the crank chamber, the piston is compressing that already in the cylinder, and while the charge in the crank chamber is being slightly compressed, the piston is receiving its impulse from the pressure due to the combustion of the charge in the cylinder. Thus a power stroke is obtained once in every revolution of the crank shaft.

Considering the two- and four-stroke cycles, applied to engines of the same size, and working under equal conditions as to speed and fuel, it would appear reasonable to suppose that a two-stroke engine should develop twice the power of the four-cycle, as it has twice as many power strokes in a given time. As an actual fact this is claimed by some interested manufacturers of two-stroke-cycle engines, but so far has not been realized in working. The reason is not very evident, but probably leakage of the charge, combined with undue dilution of the mixture with the products of combustion remaining in the cylinder, has a good deal to do with it. The compression in a two-stroke-cycle motor is not carried to so high a point as in a four-cycle, and this partly explains why they are not so economical in fuel consumption. Hence it is that although a two-stroke-cycle motor shows increased power for a given-sized cylinder, as compared with a four-cycle engine, both being operated under the same conditions as to speed, the gain is obtained at the expense of economy. It is seldom that the actual gain in power exceeds 60 per cent. over that developed by a four-cycle engine of equivalent dimensions.

The chief claim for consideration in a two-stroke-cycle engine is its undoubted simplicity. There is a total absence of valves, except the check valve in the crank chamber, and even this is abolished in one design by making the piston act to close and open a port in

communication with the carburettor. Again, there are no gears, cams, springs, etc., so that the engines are cheap to manufacture and cost little for repairs and upkeep. It would seem that this design of engine will continue to receive much attention, and it may be that it will become a powerful rival of the four-cycle motor, especially for automobile work.

In this book, however, all the formulæ relate to four-stroke-cycle engines; it is not probable that they will be superseded for some time to come by the two-stroke engine.

Symbols and Definitions.—The internal combustion engines considered in the following pages are those using petroleum spirit, having a specific gravity of 0·68 to 0·70, and with a low flash-point, as fuel. This spirit is commonly known as "petrol," and much information regarding it and its combustion will be found on pp. 48–51.

The working of an internal combustion engine is a strictly thermodynamic process, and the work done is proportional to the change in temperature. For an average case the change in temperature will be proportional to—

$$\left(\sqrt[3]{\frac{V}{V_1}} - 1\right) \quad \text{and} \quad \left(\sqrt[4]{\frac{P_1}{P}} - 1\right)^*$$

From these expressions it would appear that high compression is conducive to increased efficiency, and this assumption is verified in practice by the high thermal efficiency of the Diesel motor, in which the compression is carried to upwards of 500 lbs. per square inch. For use on an automobile, such pressures are, for the present at least, quite out of the question, and about 90 lbs. per square inch, absolute, will be found to be the practical limit. Even with this pressure some device will be necessary to allow the pressure to be partly relieved when the

* For meaning of symbols see p. 11.

motor is started, especially when the cylinder is large in diameter. The compression is limited in ordinary four- and two-cycle motors, in which the fuel is present in the cylinder during compression, by the liability to premature ignition of the charge by the rise in temperature due to compression. Somewhere about 90 to 100 lbs. per square inch will be the limit from this cause.

The compression pressure is an important factor in

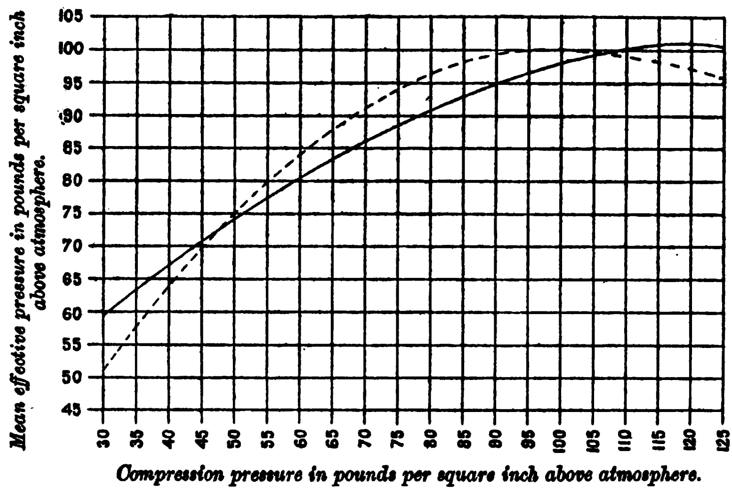


FIG. 3.

the mean effective pressure, and it is reasonable to suppose that the latter will increase in some fairly constant ratio to the rise of the former. Mr. Frederick Grover's well-known formula, M.E.P. $= 2C - 0{\cdot}01C^2$, in which $C =$ the compression pressure in pounds per square inch above atmosphere, is graphically illustrated by the dotted curve in Fig. 3. In this diagram the vertical scale gives the mean effective pressure and the horizontal scale the compression pressure, both in pounds per square inch above

atmosphere. From practical observations, the writer ventures to think that the full line in the diagram indicates results more in accordance with actual practice. It will be seen that the full line shows a fairly regular increase in the mean effective pressure as the compression pressure rises.

Much of the efficiency of a motor will depend upon the form of the combustion chamber. It should have a maximum of cubic capacity with a minimum of surface, and this condition is best fulfilled when the combustion chamber is hemi-spherical in form. Pockets, or ports, which interfere with the regular form of the combustion chamber tend to lower the heat of the burning gases and to decrease the efficiency of the motor; from which it will be gathered that those motors which have the inlet and exhaust valves on opposite sides of the cylinder, and therefore in two separate pockets, are not as efficient as they might be. The most efficient motor, other things being equal, will be that in which the valves open directly into the combustion chamber without the intervention of any ports or passages.

Before a motor can be designed, certain data must be obtained, and some factors assumed. Usually the brake horse-power, number of cylinders, and the number of revolutions are given. The compression pressure may be assumed, but should always have as high a value as possible. The number of cylinders will be governed by the space at command and on the permissible amount of vibration. The speed of the motor is also to a certain extent decided by the space available, since the power is proportional to the speed, other things being equal, and within the permissible limits for the piston speed.

In the formulæ used for determining the dimensions of the cylinder and combustion chamber, the symbols employed have the following meanings:—

Let P = the pressure existing in the cylinder at the commencement of the compression stroke, in pounds per square inch absolute.

P_1 = the compression pressure, in pounds per square inch absolute.

P_2 = the maximum pressure due to combustion, in pounds per square inch absolute.

P_3 = the pressure at the termination of the expansion stroke, in pounds per square inch absolute.

A = the cylinder area in square inches.

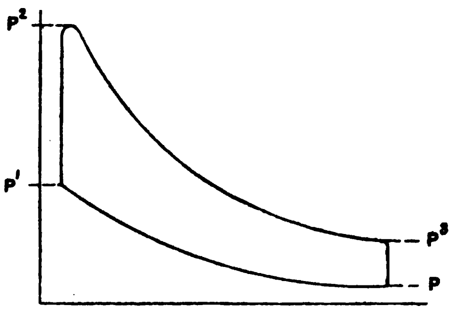


FIG. 4.

V = the total cylinder length, with the piston at the termination of its out-stroke, in feet.

V_1 = the total length of the compression space, with the piston at the termination of its in-stroke, in feet.

T = the absolute temperature of the charge at the commencement of the compression stroke.

T_1 = the absolute temperature of the charge at the termination of the compression stroke.

N = the number of revolutions per minute.

The symbols P, P_1, P_2, P_3, are shown graphically in Fig. 4, and V and V_1 in Fig. 5.

Cylinder Design.—If W represents the work done in the cylinder of an internal combustion engine, in foot-pounds per working stroke, then—

(1) $$W = 110\ AV\left[\left(\frac{V}{V_1}\right)^{\frac{1}{3}} - 1\right]$$

or—

(2) $$W = 110\ AV\left[\left(\frac{P_1}{P}\right)^{\frac{1}{3}} - 1\right] \quad \text{(Stoddard)}$$

To be of practical use we require these expressions modified for the indicated horse-power. This we may obtain by multiplying by the number of impulses per

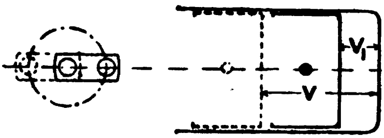


FIG. 5.

minute, equal to half the number of revolutions per minute, and dividing by 33,000; thus—

(3) $$\text{I.H.P.} = \frac{110\ AV\left[\left(\frac{V}{V_1}\right)^{\frac{1}{3}} - 1\right]\frac{1}{2}N}{33000}$$

or—

(4) $$\text{I.H.P.} = \frac{110\ AV\left[\left(\frac{P_1}{P}\right)^{\frac{1}{3}} - 1\right]\frac{1}{2}N}{33000}$$

and after reducing these two expressions become—

(5) $$\text{I.H.P.} = \frac{AV\left[\left(\frac{V}{V_1}\right)^{\frac{1}{3}} - 1\right]N}{600}.$$

or—

(6) $$\text{I.H.P.} = \frac{AV\left[\left(\frac{P_1}{P}\right)^{\frac{1}{4}} - 1\right]N}{600}$$

These last two expressions may be reduced to the following forms, which are more convenient in use. The table of roots, p. 139, will be useful in this connection.

(7) $$AV = \frac{600 \times \text{I.H.P.}}{\left(\sqrt[3]{\frac{V}{V_1}} - 1\right)N}$$

or—

(8) $$AV = \frac{600 \times \text{I.H.P.}}{\left(\sqrt[4]{\frac{P_1}{P}} - 1\right)N}$$

In all these formulæ we have only considered the indicated horse-power. The brake horse-power will depend upon the mechanical efficiency of the engine, which for the larger sizes, say 15 H.P. and over, may be safely assumed at 80 per cent., and for smaller engines 70 per cent. Put in other words, the brake horse-power will be obtained by multiplying the indicated horse-power by 0·8 and 0·7 for large and small motors respectively.

To illustrate the use of the formulæ, we will take as an example a motor having two cylinders capable of developing 12 brake horse-power at a normal speed of 900 revolutions per minute. We will assume the compression to be 70 lbs. per square inch absolute, *i.e.* 55 lbs. per square inch above atmosphere. Each of the two cylinders must develop half the total power—that is, 6 brake horse-power—and for a motor of this size we shall be safe in taking the mechanical efficiency at 70 per cent.

Hence each cylinder must be capable of developing $\frac{6}{0.7}$ $= 8.57$ indicated horse-power.

Substituting these known values for their symbols in formula 8, we have—

$$AV = \frac{600 \times 8.57}{(\sqrt[4]{\frac{70}{14}} - 1)900} = \frac{2 \times 8.57}{(\sqrt[4]{5} - 1)3} = \frac{17.14}{1.485} = \text{say } 11.5 \text{ ft.}$$

Having the value of AV in feet, we may reduce it to inches by multiplying by 12, and hence—

$$AV = 11.5 \times 12 = 138 \text{ inches}$$

We may now assume the value of either of the factors A or V, and obtain that of the other by calculating. It is convenient to assume the value of A, and in this case we will take it as 15·9 square inches, *i.e.* the area of a cylinder $4\frac{1}{2}$ inches diameter. Expressing this as an equation, we have—

$$15.9\ V = 138$$
$$\therefore V = 8.68 \text{ inches}$$

This gives us the total length of the cylinder with the piston at the end of its out-stroke. By subtracting from this the length of cylinder necessary for the compression space, corresponding to V_1, we shall obtain the stroke. To simplify this and eliminate calculations, the diagram, Fig. 6, has been prepared. To obtain the length of the compression space it is necessary to ascertain the value of the ratio $\frac{V_1}{V}$ corresponding to 70 lbs. per square inch absolute. In the diagram, Fig. 6, commencing at the point indicating 70 lbs. on the right-hand vertical scale, draw a horizontal line to cut the curve AB, and from the point of intersection drop a perpendicular line to cut

the base line. This will fall on the point marked ·3. The value of the ratio $\frac{V_1}{V}$ for 70 lbs. compression absolute is therefore 0·3. By multiplying the value of V by this

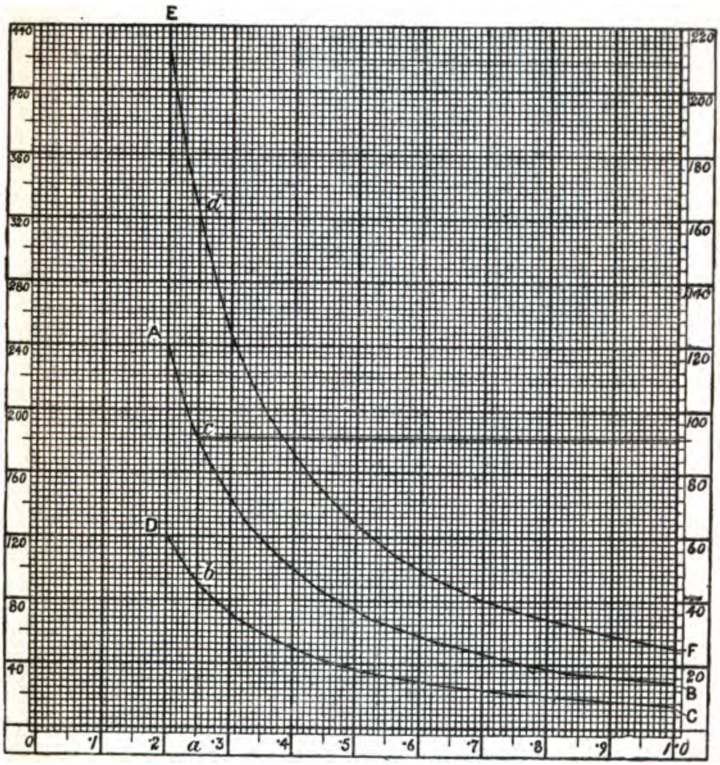


FIG. 6.

figure, we obtain the length of the compression space; thus—

$$8·68 \times 0·3 = 2·6 \text{ inches}$$

The compression space (V_1) is therefore = 4½ inches

diameter × 2·6 inches long, and the stroke of the piston will be 8·68 − 2·6 = 6·08 inches. It will be sufficiently accurate for all practical purposes if we take the length of stroke as 6 inches, and the length of the compression space as 2⅝ inches. In other words, the length of the cylinder denoted by V should contain 138 cubic inches, and V_1 42 cubic inches. The diagram, Fig. 6, will be of considerable assistance to a motor designer, as from it may be obtained, without calculation, the ratio of the cylinder and compression-space volumes for a given compression, the compression pressure corresponding to a given ratio of volumes, the approximate maximum explosion pressure, and the theoretical indicator diagram. In the diagram the vertical scales represent pressures in pounds per square inch absolute, *i.e.* gauge pressure plus atmospheric pressure, which in this instance has been taken as 14·5 lbs. per square inch. The horizontal scale at the bottom represents the total cylinder length (V), and is divided into tenths and hundredths. The method of reading the diagram is as follows: The curve AB shows the relation between volume and pressure during the compression stroke, and is to be read by the scale on the left. The curve CD has the same significance as AB, except that it is to be read by the left-hand scale of pressures. The curve EF represents the relation between volume and pressure after the charge has been ignited, or, briefly, is the expansion line.

For example, suppose we have a motor with a ratio of volumes ($V_1 : V$) of 1 to 4, or 0·25. What will the compression and maximum explosion pressures be, assuming that a correct mixture of air and gas is employed as fuel?

At the point *a* on the horizontal scale, corresponding to the given ratio of volumes, $\frac{V_1}{V} = \frac{1}{4}$, or 0·25, erect a perpendicular *abc*, cutting the curve AB at *c*, and from

c draw a horizontal line to the scale on the right. This will be found to touch the scale at a point corresponding to about 90 lbs., which is therefore the absolute compression pressure. To find the maximum pressure due to combustion, continue the line *abc* to cut the curve EF at *d*, and from *d* draw a horizontal line, *de*, to the left-hand scale, and where it touches the scale the maximum pressure may be read. In this instance it is about 320 lbs. absolute, or 305 above atmosphere.

The theoretical indicator diagram is to be read entirely by the left-hand scale, and is represented by the curve C*b*, compression line; *bd*, explosion line; curve *d*F, expansion line; and FC, exhaust line. The area enclosed by these curves and lines represents the indicated work of the engine per working stroke.

From the diagram, Fig. 3, the mean effective pressure may be read directly. In this figure the horizontal scale at the bottom represents the compression pressure in pounds per square inch above atmosphere, and the vertical scale on the left the mean effective pressure, also in pounds per square inch above atmosphere. As mentioned previously, this curve has not been obtained solely from theoretical considerations, but has been deduced mainly from recorded data. It is not put forward as being exact, but as a close approximation to actual practical conditions. The compression pressure is an important factor in the mean effective pressure, inasmuch as a reduction of one pound in the compression pressure will make a difference of nearly 10 per cent. in the mean effective pressure. To make the relation between the pressures and volumes quite clear at a glance, they may be expressed in the form of equations; thus—

(9) $$P_1 = P\frac{V}{V_1}\sqrt[3]{\frac{V}{V_1}}$$

$$(10) \qquad V_1 = V\sqrt{\frac{P}{P_1}\sqrt{\frac{P}{P_1}}}$$

both of which are convenient to use with ordinary tables of roots, and give results as accurate as can be expected from any general formulæ.

When a charge is drawn into the cylinder it will be heated, and will then expand. Consequently the actual amount of combustible mixture taken into the cylinder will always be less than the theoretical quantity. The heating is mainly caused by the hot residual gases of the previous stroke. We may therefore assume that the rise in temperature from this cause will be approximately equal to the ratio of the volume of the hot gases to the volume of the charge drawn in, or to $\frac{V_1}{V - V_1}$.

Assuming the temperature of the air to be 60° Fahr., *i.e.* 520° Fahr. absolute, and adding to this the above ratio, multiplied by a constant (280), which gives results corresponding to recorded data, we shall have—

$$(11) \qquad T = 280\left(\frac{V_1}{V - V_1}\right) + 520$$

By multiplying the results obtained from equation 11 by the expression $\sqrt[3]{\frac{V}{V_1}}$ we shall have the temperature at the end of the compression stroke; thus—

$$(12) \qquad T_1 = T\sqrt[3]{\frac{V}{V_1}}$$

In the following table the values of T and T_1 have been calculated from equations 11 and 12, and the volume ratios to which they correspond will be found in the first column:—

TABLE 1.

$\frac{V_1}{V}$	T	T_1
0·5	800	1008
0·475	773	990
0·45	749	977
0·425	727	967
0·4	707	959
0·375	688	955
0·35	671	952
0·325	655	953
0·3	640	956
0·275	627	964
0·25	613	973
0·225	601	988
0·2	589	1008

Now the average value of T_1, as shown by the last column, is 973° Fahr. absolute, and this average varies less than 4 per cent. from the two extremes. Therefore we shall not introduce any serious errors into our calculations if we assume the temperature of compression as constant and equal to 973° Fahr. absolute.

With a rich mixture, and with all conditions favourable, the temperature will rise to somewhere about 3400° Fahr. absolute, on ignition. The exact temperature attained when the charge is ignited is at the present time still a matter for some conjecture, but the above value will be close enough for the purpose of these calculations. The pressure rises in proportion to the temperature, so that by multiplying the compression pressure by the ratio $\frac{3400}{973}$ we shall have the maximum pressure due to combustion. Hence—

(13) $$P_2 = P_1 \frac{3400}{973} = 3{\cdot}5 P_1 \text{ nearly}$$

If it is correct to assume the same law for expansion

as for compression, the pressure at the termination of the expansion stroke should be obtained by the following expression:—

$$(14) \quad P_3 = 3{\cdot}5P = 3{\cdot}5 \times 14{\cdot}7 = 51{\cdot}45 \text{ lbs. absolute}$$

There is some doubt as to the absolute accuracy of this last expression, as the same law does not quite answer for compression and expansion, but the actual results approximate very closely to recorded data, so that the formula may be allowed to stand as a convenient approximation.

The ratio between the cylinder diameter and the length of stroke is mainly determined by the piston speed allowed. This varies between wide limits in actual practice, ranging from 600 to as much as 1800 feet per minute. The most economical piston speeds are, for vertical motors, 800 feet per minute, and for horizontal motors 700 feet per minute. Where space is limited, and a larger output from a given-sized engine is required, these speeds may be somewhat exceeded, with a corresponding loss of efficiency.

In small motors the thickness of the cylinder walls is not so much a matter for calculation, but is rather determined by the requirements of the foundry. It is most unlikely that small cylinders will be cast so thin as to be unsafe, but with cylinders of 4-inch diameter and upwards it is advisable to calculate the thickness. A convenient formula for rapidly approximating the thickness of the walls is—

$$(15) \qquad K = 0{\cdot}075D$$

where K = the thickness of the cylinder walls in inches, and D = the diameter of the cylinder bore in inches. If we wish to calculate the thickness, taking into account the safe stress per square inch to be allowed for the

material of which the cylinder is composed, the following equation may be employed:—

$$(16) \qquad K = \frac{P_1 D}{2f}$$

in which K = the thickness of the cylinder walls in inches, D = the diameter of the cylinder bore in inches, P_1 = the absolute compression pressure, and f = the safe working stress in pounds per square inch. If we take the maximum pressure as 3·5 times the compression pressure, and the safe stress at 3500 lbs., the expression becomes—

$$(17) \qquad K = \frac{3 \cdot 5 P_1 D}{7000}$$

which after reducing will be—

$$(18) \qquad K = \frac{P_1 D}{2000}$$

There does not appear to be any need for a formula for the thickness of the water-jacket walls, as there is no stress to speak of to be resisted by them. If made thick enough to obtain good castings, the jacket walls will be quite strong enough for all other purposes. If a rule is required for the sake of uniformity of design, then the jacket wall may be made half the thickness of the cylinder wall.

The water space around the cylinder should bear some relation to the thickness of the cylinder wall, and a ratio which gives good results is to make this space 1·5 times the thickness, or 1·5 K. For small cylinders the space will probably be determined by the ability of the moulder who makes the casting, rather than by the formula. The limit for the water space from this cause will be about $\frac{3}{8}$ inch. In designing the cylinder and its water jacket,

care should be taken to avoid all pockets in which air or steam may collect and prevent the water coming in contact with the cylinder walls, as this would tend to cause unequal cooling, with perhaps serious results. The importance of keeping the cylinder uniform in shape, and without pockets, has been mentioned on p. 10.

In motors which have more than one cylinder cast integral with the water jacket, it is good practice to arrange for a water space between the cylinders. This not only ensures more equal cooling effect, but, by the thickness of the metal being made more uniform, will tend to prevent sponginess in the castings. All water-jacketed cylinders should be tested by water pressure to at least 50 lbs. per square inch, inside the jacket space, to make sure that the metal is sound, and this should be done after the cylinder bore has been machined. If there are very small cracks in the jacket, the casting need not be rejected, as by filling the jacket with a solution of salammoniac the cracks can be rusted up. If, however, there are the slightest signs of water percolating into the cylinder or combustion chamber, the casting should be replaced with a sound one. The sudden and severe stresses to which the cylinder, and especially the combustion chamber, are subjected render the use of the rusting-up process an unsafe remedy.

The valve chamber should always be well water jacketed, especially around the exhaust valve. The inlet valve can generally be left unjacketed, as the rush of the cool mixture past this valve will, as a rule, keep this part cool enough. In this connection it may be noted that the sparking plug should always be located in such a position that the cool incoming charge will impinge upon the sparking points, and thus prevent them becoming hot enough to cause premature and irregular ignition.

Valves.—Until recently it was a very common

occurrence to find the valves and valve ports considerably smaller than they should have been. The importance of ample area for the valves and valve passages may be gauged from the fact that for each pound reduction of pressure below atmosphere at the commencement of the compression stroke, the power of the engine will be lowered about 10 per cent. A short lift to the valves allows of their closing in a shorter time than when the lift is high, and for this reason the diameter should be kept as large as can be conveniently allowed. The main factor in determining the areas of the valves and passages is the speed at which the gases will pass through them. Hence the size of the valves will depend upon the area of the piston and its speed in feet per minute.

For the inlet valve and port, the area should be such that the speed of the gases will not exceed 100 feet per second, and for the exhaust valve 85 feet per second. If the exhaust gases were expelled from the cylinder at atmospheric pressure the allowable speed could be the same as for the incoming charge; but as at the moment of release the pressure is never much less than 25 lbs. above atmosphere, and may be as high as 51 lbs. (see formula 14, p. 20), the lower speed is taken.

A common rule for the valve dimensions is to make the inlet-valve area one-twelfth and the exhaust-valve area one-tenth the area of the cylinder. For roughly approximating the sizes of the valves this rule answers fairly well, but to obtain the best results the following formulæ should be employed:—

Let S = the piston speed in feet per minute.
A = the area of the cylinder in square inches.
I = the area of the inlet valve in square inches.
E = the area of the exhaust valve in square inches.

Then—

$$(19) \qquad I = \frac{AS}{6000}$$

$$(20) \qquad E = \frac{AS}{5000}$$

The angle of the valve seatings should be 45° to the vertical axis of the valve stem. If made more acute, there is some risk of the valve sticking in the seat; if much flatter, particles of carbonaceous matter may adhere, and so prevent the valve closing properly. The width of the actual seating may be about equal to half the thickness of the head of the valve, or 0·05 times the diameter of the valve opening. The wider the seating, in reason, the longer the valve will work without it being necessary to re-grind the seating. Also the pitting and erosion due to the rapid passage of the hot products of combustion is much more pronounced with narrow than with wide seats. The valves themselves are usually made of one piece of mild steel, but in the case of large valves the head is sometimes made of cast iron, or even nickel alloy, which is screwed and riveted to a mild-steel stem. This is claimed to make a more durable valve than one constructed entirely of steel, but the writer's observations go to show that the greater amount of pitting and erosion take place on the valve seat, and that a mild-steel valve head is practically as good as one of cast iron. It might be expected that nickel steel would give the best all-round results as a material for exhaust valves, but the writer has no data on this point. When the head of the valve is made separate from the stem, there will be a possibility of the head becoming loose on the stem, which is entirely avoided by the one-piece valve.

It is of the utmost importance that the guide, in which the stem of the valve works, should be perfectly concentric

with the valve seat. The practice of making the valve-stem guide separate and screwing it into the valve box is to be deprecated, as it usually results in the guide being eccentric with the seat. Even if made true to commence with, the expansion and contraction due to the changes in temperature will, in the majority of cases, cause the guide to become eccentric with the seat sooner or later. If cast integral with the valve box, the guide can be made true with the seat once for all, and will remain so. If, however, the valve box is insufficiently water jacketed, or is cooled on one side more than another, there will be considerable risk of the valve-stem guide being warped when the valve box is heated.

Next to having the valves the correct size, the matter of timing their operation takes an important position. The exact moment at which the exhaust valve should open depends, for the most part, on the piston speed. A motor running with a high piston speed will require to have the exhaust valve opened considerably earlier in the cycle than when the piston speed is low. There is not much data available on this point, but as a guide it may be stated that with a motor having a piston speed of 700 feet per minute, the exhaust valve should commence to open when the piston has completed eight-tenths of its stroke, and close when the piston has started on its suction stroke, and *not* exactly at the dead centre. The reason why the valve should be late in closing, although this is contrary to usual practice, is that with a high piston speed the products of combustion will not have all escaped from the cylinder at the termination of the exhaust stroke. The exact amount by which the closing of the exhaust valve should be delayed will best be determined by experiment. At slow piston speeds of, say, 500 feet per minute, the exhaust valve may be closed exactly at the dead centre. The writer has observed a decided improvement in power,

in the case of high-speed motors, when the closing of the exhaust valve has been thus delayed, hence it is reasonable to assume that the products of combustion are not entirely expelled by the time the piston has finished its in-stroke. From consideration of this point the writer has for some time been of the opinion that the mechanically operated inlet valve, as applied to *high-speed* engines, is a mistake. The usual practice is to open the inlet valve immediately the exhaust valve closes, without reference to the pressure existing in the cylinder. Thus if the pressure within the cylinder is above atmospheric at the moment the inlet valve is opened, in place of a fresh mixture flowing into the combustion chamber, there will be a rush of the residual, burnt gases into the carburettor. Before any fresh mixture of air and gas can be taken into the cylinder, these burnt gases must be drawn back through the inlet valve, and the greater the piston speed, the more will this effect obtain. With a motor required to run at a low piston speed, the mechanically operated inlet valve undoubtedly gives better efficiency than the automatic valve, and in point of fact the chief claim made by the advocates of the mechanical valve is that it enables the motor to be run at a much slower speed than when it is equipped with automatic inlet valves.

The most rational method of overcoming the defects of the automatic and mechanically operated inlet valves would seem to be that in which the moment at which the valve opens should be determined by the pressure existing in the combustion chamber, as in the valve gear invented by Mr. R. E. Phillips, M.I.M.E., where the pressure of the spring which holds the inlet valve on its seat is relieved a short time before the completion of the exhaust stroke, and the inlet valve is kept closed by the pressure of the gases in the cylinder. When the pressure in the combustion chamber falls to that of the atmosphere the inlet valve is

free to open, either by its own weight, if inverted, or by the suction effect of the piston, without the restraining influence of any spring. A light spring may even be used to assist the valve in opening. Prompt closing of the valve is ensured by the valve spring being allowed to again resume its function. With automatic inlet valves the spring tension is a matter requiring careful adjustment. If too strong, the valve will only open, and remain open, while the pressure in the cylinder is below that of the atmosphere to an extent depending upon the strength of the spring, resulting in small charges and a consequent lowering of the compression pressure. With a weak spring the valve will open with a very slight vacuum in the combustion chamber, and thus full charges will be assured, but the closing of the valve may be so delayed that the greater portion of the charge will be returned into the carburettor. At the best only a compromise is possible, and the general tendency is towards using a fairly strong spring, and rightly so, as the lesser of two evils. The time taken by the inlet valve in closing, especially with high-speed motors, is important. This time may be calculated from the following formula:—

$$(21) \qquad S = 0{\cdot}0721\sqrt{\frac{LW}{M}}$$

in which S = the time in seconds, L = the lift of the valve in inches, W = the weight of the valve, and M = the average pressure exerted by the spring. W and M must both be taken in the same units, either ounces or pounds. Taking, for example, an engine running at 800 revolutions per minute, with a valve weighing 6 ounces, and having a lift of, say, $\frac{11}{32} = 0{\cdot}34375$ inches. Assuming the spring selected to have an average tension of 12 ounces, and substituting known values in the formula 21, we have—

$$S = 0{\cdot}0721\sqrt{\frac{0{\cdot}344 \times 6}{12}} = \text{about } 0{\cdot}03 \text{ second}$$

At 800 revolutions per minute the engine would make one revolution in $\frac{60}{800} = 0{\cdot}075$ second, or one stroke in 0·038 second; that is, the engine would make nearly one complete stroke while the valve is closing. Evidently a much stronger spring is required. To calculate the size of spring to be used the formulæ given by Professor Unwin

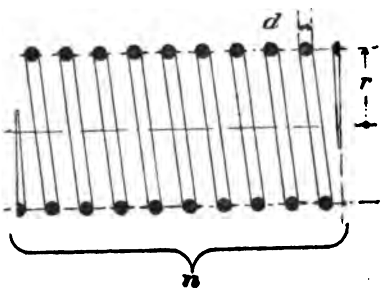


FIG. 7.

will be found of great utility. For the force required to compress, or extend, the spring, we have—

$$(22) \qquad F = \frac{3250000d^4}{nr^3}$$

in which F = the force necessary to compress (or extend) the spring one inch, in ounces; d = the diameter of the wire in inches; r = the mean radius of the coil in inches; and n = the number of coils. These proportions are graphically illustrated in Fig. 7.

The formula for the safe working load on the spring, in pounds, is—

$$(23) \qquad F = \frac{10000d^3}{r}$$

the notation being the same as for formula 22. To facilitate the application of these two formulæ, the following table has been calculated:—

TABLE 2.

No. of wire, B.W.G.	Diameter of wire in inches.	$3,250,000d^4$ ounces.	$10,000d^4$ pounds.
6	0·203	5519·0	83·4
7	0·18	3412·0	58·32
8	0·165	2409·0	44·92
9	0·148	1560·0	32·4
10	0·134	1048·0	24·0
11	0·12	674·0	17·28
12	0·109	458·8	12·96
13	0·095	264·7	8·58
14	0·083	154·3	5·72
15	0·072	87·34	3·74
16	0·065	58·02	2·74
17	0·058	36·78	2·0
18	0·04	18·74	1·17
19	0·042	10·12	0·74
20	0·035	4·88	0·428
21	0·032	3·41	0·328
22	0·028	2·0	0·22
23	0·025	1·27	0·154
24	0·022	0·7614	0·106
25	0·02	0·52	0·08
26	0·018	0·3412	0·058
27	0·016	0·213	0·041
28	0·014	0·125	0·028
29	0·013	0·093	0·022
30	0·012	0·0674	0·014

When the force required to compress, or extend, the spring 1 inch has been found, the force necessary to compress it more or less can be ascertained by simple proportion.

In the example selected above we found that a spring with an average tension of 12 ozs. was much too weak to ensure prompt closing of the valve. Suppose we decide to try a 19-lb. spring to increase the speed of closing.

We shall have for the size of wire, from formula 23, assuming a mean radius of $\frac{1}{4}$ inch—

$$19 = \frac{10000d^3}{0{\cdot}25}, \therefore 4{\cdot}75 = 10000d^3$$

Looking in the fourth column of Table 2, the nearest (higher) number to 4·75 is 5·72, corresponding to a No. 14 gauge wire, which will therefore be strong enough, provided the tension does not much exceed 19 lbs. Suppose we have room enough for 32 coils; substituting known values in formula 22, we have for the force necessary to compress the spring 1 inch—

$$F = \frac{3250000d^4}{32 \times 0{\cdot}25^3} \therefore 0{\cdot}5F = 3250000d^4$$

From the third column in Table 2 we find that the value of the second member of this equation, for a 14 gauge wire, is 154·3, and therefore the force required to compress our spring 1 inch will be—

$$154 \times 2 = 308 \text{ ozs., or } 19{\cdot}25 \text{ lbs.}$$

With this spring the valve will close in 0·0065 second, that is, in less than one-fifth of a stroke. The actual force exerted by the spring on the valve will be practically in proportion to the lift, or, as we have assumed a lift of $\frac{11}{32}$ inch, it will be equal to $19 \times 0{\cdot}34375 = 6{\cdot}5$ lbs.

The arrangement of the valves differs considerably, and in some designs efficiency is sacrificed to obtain a motor symmetrical in appearance. It should be the aim of the designer to reduce, as much as possible, the length of the passage leading from the combustion chamber to the valve box. Long passages, especially if much curved, tend to cool the gases, and so lower the thermal efficiency of the motor. Other things being equal, the valves will be in

the best position when they open directly into the cylinder. The Maudslay motor is an excellent example of correct placing of the valves (Fig. 8).

In designing the valve box there are two points which should not be lost sight of. One is that the exhaust-valve seat should be sunk below the level of the port leading to the combustion chamber, so that the products of

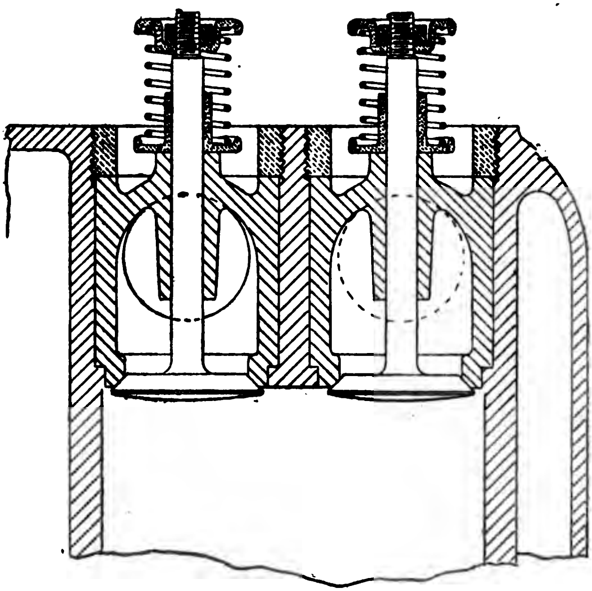


Fig. 8.

combustion do not impinge upon one side of the valve, but shall be compelled to flow equally all round it. If the gases strike the valve on one side only, it, and the seating, will be pitted and burnt in one part more than another, and regrinding will seldom effect a cure for this. The annular recess around the valve should provide ample area for the passage of the gases.

The second point is to make sure that there is sufficient room around the valve head, when lifted, for the easy passage of the gases. The annular space between the periphery of either the inlet or exhaust valve and the interior wall of the valve chamber should be about one-fifth greater than the actual area of the valve opening.

It is as well to have a standard of proportions for the valves, and those given in Fig. 9 will be found to work out well in practice for exhaust valves. The same proportions may well be used for mechanically operated inlet

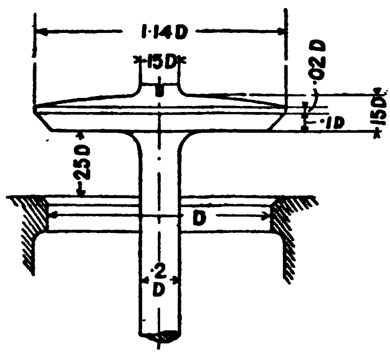


FIG. 9.

valves, but for automatic inlet valves they may be made about 15 per cent. less. It is convenient to make the inlet and exhaust valves of the same size, and interchangeable, when both are operated mechanically, calculating the area of both by formula 20, *i.e.* the inlet valve should be as large as the exhaust requires to be, and not *vice versâ*.

To provide ample surface to resist wear, and to prevent leakage as far as is reasonably necessary, the exhaust-valve stem guide should not be less than eight times the diameter of the stem in length, and will be all the better if made

10 diameters long. As the conditions of working are much less severe in the case of the inlet valve, the stem guide for this may be safely made 6 diameters long.

The Piston.—There appears to be considerable variation in the proportions adopted in the design of pistons; in some cases they are made as much as two and a half times the diameter in length, while in others the length and diameter are made equal. A standard may be adopted for the length, based on consideration of the wearing surface necessary, which will be found to agree with the average proportions used by the best-known makers. The piston of an internal-combustion motor has not only to transmit the energy of the "explosion" through the medium of the connecting rod to the crank shaft, but has also to act as a guide, and receive the angular thrust of the connecting rod.

Fig. 10 gives suggested proportions for the piston, all the dimensions being based on the diameter as a unit with the exception of the gudgeon pin, which is best designed from consideration of the stresses it has to bear. At least three piston rings should be used on all but the smallest pistons, such as motor-bicycle engines, up to 2½ inches diameter, and for pistons larger than 3½ inches diameter four rings are advisable. The use of an extra ring near the outer end of the piston is of doubtful advantage.

The proper fit of the piston in the cylinder is a matter requiring some skill to accomplish satisfactorily. A point very often overlooked is the expansion of the end plate, or piston head, owing to its being in contact with the burning charge during the power stroke. From this cause the closed end of the piston will expand a sensibly greater amount than the open end; hence, when cold, the form of the piston should be such that the unequal expansion will be accommodated when the piston is heated. It is by no

means uncommon to find that a motor will give more power, and run more sweetly, after having been used some time than when quite new. This is probably due to the piston having been originally turned to fit the cylinder closely for the whole of its length, so that, when it is heated in running, binding takes place to a greater or lesser extent around the closed end, until by continued

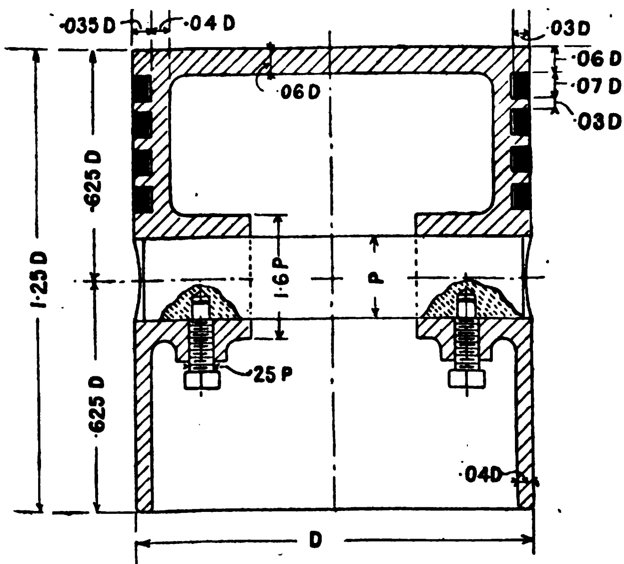


FIG. 10.

running the piston wears on the parts which bear hardest against the cylinder walls. The writer's method is to make the piston perceptibly smaller at the closed end before any running is attempted, to compensate for expansion. The allowance usually required is quite small, but the improvement in the running of the motor is very marked. About 0·01 inch on the diameter for each inch

diameter of the piston will be found a good working allowance. The reduction in diameter can be gradual, tapering from the standard diameter near the gudgeon pin down to the full reduction at the closed end. Another method, which gives equally good results, is to make the diameter slightly less on each of the belts between the piston rings, commencing with the standard diameter at the gudgeon pin, and dividing the total amount allowed for the reduction into equally proportioned steps between the rings. When a number of motors are to be made of one size it will be advisable to experiment with the first of the series till the piston bears equally along its whole length when hot, and to then carefully take the dimensions at various points with a micrometer, and to enter them on the piston drawing.

Piston Rings.—To obtain a practically equal amount of pressure over the whole circumference of the piston rings, they should be made thicker on one side than the other, the cut, or split, being made at the thinnest part. The outside should be turned a dead fit to the cylinder bore after the ring has been cut, and with the opening, or slit, quite closed. The practice of turning the rings "just a little" larger than the cylinder, making a plain diagonal saw-cut in them, and then springing them into the cylinder, cannot be too strongly condemned. Such rings are necessarily more or less oval in form, and the chances are that before they wear to a circular shape the cylinder bore will be worn somewhat oval, in which case the only remedy will be reboring. Also there can be no certainty that the joints of such rings will be close, and if they are not there is sure to be some leakage past them. The Davy-Robertson rings have much to recommend them. They are turned a dead fit to the cylinder bore, and are parallel in thickness all round. The necessary spring is obtained by hammering them on their inner surface, the

ring being placed within a close-fitting die the while. The force of the hammer blows is graduated, being at a maximum opposite the joint, and at a minimum just at the joint. By this process the rings have perfectly uniform spring imparted to them. Moreover, the ring being of equal thickness all round, there is less chance of leakage by the gases passing behind the rings, as they fill the grooves in the piston more completely than eccentric rings.

Fig. 11 shows the proportions for eccentric rings, the

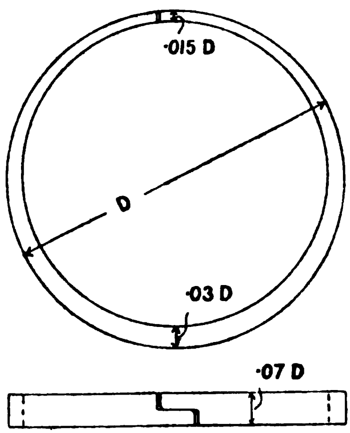


FIG. 11.

unit being the cylinder diameter, and also illustrates the best form of joint. Hard cast iron is the best material for piston rings; if made of steel, there is considerable risk of the cylinder walls being scored, and also the spring of cast-iron rings is superior and more lasting than that of steel rings.

Gudgeon Pin.—The maximum pressure allowable on the gudgeon pin is 800 lbs. per square inch. This figure does not apply to the actual strength of the pin, but

rather to the adaptability of the bearing surfaces to retain the lubricant. In determining the size of the gudgeon pin the maximum pressure in the cylinder is used as a factor, and this may be obtained from the formula—

$$(24) \qquad P_2 = 50A\frac{V}{V_1}\sqrt[3]{\frac{V}{V_1}}$$

in which the meaning of the symbols is as given on p. 11. The maximum pressure may also be read from the diagram, Fig. 6, p. 15. The diameter of the gudgeon pin may be found from the following formula—

$$(25) \qquad d = 0{\cdot}06\sqrt[3]{P_2LD^2}$$

in which d = the diameter of the gudgeon pin in inches, D = the diameter of the cylinder in inches, L = the crank radius in inches, and P_2 = the maximum pressure in the cylinder.

Crank Shaft.—The allowable pressure on the journals of the crank shaft and on the crank pin should not exceed 400 lbs. per square inch. For the diameter of the crank shaft the following formula will be found to give liberal dimensions; but considering the great stresses to which the crank of an automobile engine is subjected, the size obtained will not be greater than is required to provide a good factor of safety—

$$(26) \qquad d = 0{\cdot}06D\sqrt[3]{P_2}$$

where d = the diameter of the crank shaft, D = the diameter of the cylinder, and P_2 the maximum pressure in the cylinder in pounds per square inch absolute. Formula 26 is suitable for cases where the length of the journal does not exceed 1·5 times the diameter. When the ratio of the length to the diameter is greater than this, the following may be employed—

$$d = 0{\cdot}05\sqrt[3]{P_2LD^2} \tag{27}$$

in which L = the length of the journal, and the other factors are as above. In both these formulæ the diameters and the length are to be expressed in inches.

If we wish to first assume the diameter of the journal, we can obtain the length from—

$$L = \frac{Ap}{400d} \tag{28}$$

in which the factors are L = length of the journal in inches, A = the area of the cylinder in square inches, d = the diameter of the shaft or crank pin in inches, and

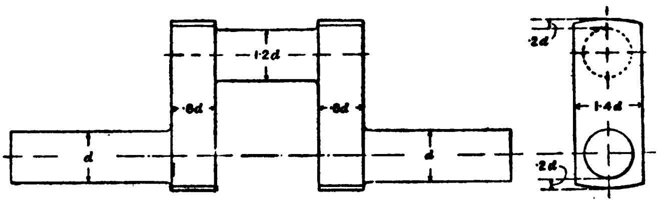


FIG. 12.

p = the mean effective pressure in pounds per square inch above atmosphere. This formula allows for a pressure not exceeding 400 lbs. per square inch on the projected area of the bearing surface.

Suggested proportions for single-throw crank shafts are given in Fig. 12, based upon the diameter of the shaft as a unit. For crank shafts with more than one throw, the formulæ given above may be used, but the diameters so obtained should be increased by 10 per cent. in the case of two-throw shafts, and by 15 per cent. for four-throw shafts. If there is a bearing between each throw, then the webs will be strong enough if made to the proportions

in Fig. 12. Sometimes it is necessary to do without a centre bearing in a two-throw shaft, owing to lack of space. In this case the centre web should be made 1·3 to 1·5 times the thickness of the outer webs.

Connecting Rods.—The connecting rod used in a petrol motor is usually of the marine type, so far as the big end is concerned, the small end being provided with a simple bush, and non-adjustable. Provided there is sufficient bearing surface, ample lubrication, and suitable materials are employed, there is very little gain in making the small end of the rod adjustable. Under proper conditions the engine may be run for a long time before the gudgeon pin becomes loose in the bush, and when this does occur it is cheaper to renew the bush than to spend time in readjusting a small end bearing of the usual type.

The section of the rod itself varies in different engines, but the most usual section is rectangular. Circular cross-sectioned rods are used to some extent, and they are somewhat cheaper to machine, being entirely finished in the lathe. For rods of circular cross-section, the following formulæ will be convenient for arriving at the *mean* diameter—

$$(29) \qquad d = 0{\cdot}09\sqrt{\mathrm{LD}}\sqrt[8]{\frac{\mathrm{V}}{\mathrm{V_1}}}$$

$$(30) \qquad d = 0{\cdot}09\sqrt{\mathrm{LD}}\sqrt[4]{\frac{\mathrm{P^1}}{\mathrm{P}}}$$

in which L = the length of the connecting rod in inches, from centre to centre; D = the diameter of the cylinder in inches; and P, P_1, V, V_1, as on p. 11.

For a rod of rectangular cross-section the thickness may be 0·45 of the diameter as found by the above formula, and the width 2·5 times the thickness. All the

above data apply to rods made from mild-steel forgings. If malleable cast iron be used, the dimensions should be increased in inverse proportion to the relative strengths of the material as compared with that of mild steel.

It is usual to rely on the oil splashed about the crank case for the lubrication of both ends of the connecting rod, but of late there have been motors constructed wherein the lubrication is effected by pumping the oil under pressure to the bearings through small pipes, as in the well-known Belliss and Morcom high-speed steam engine. The remarkable freedom from wear in the Belliss engine would seem to promise that by the same means petrol motors may be made much more durable than at the present time. Also some anxiety would be saved the operator. By providing suitable ducts the pressure system could be made to lubricate the piston as well. Another point in favour of forced lubrication is that the small oil-ways and pipes are not so liable to become stopped up as when the oil is merely allowed to run through them by gravity.

The Flywheel.—As compared with steam engines of equal power, petrol motors, especially when single cylindered, require very heavy flywheels. This is, of course, due to the great proportion of idle strokes made by the piston. Motors having three, four, or more cylinders may have flywheels considerably lighter than when only one cylinder is used, owing to the greater regularity of the turning moment. In the formulæ given below, this point has received attention by the provision of a factor representing the proportion of impulses to the revolutions per minute. Other things being equal, a flywheel of large diameter will be more efficient than a small one of equal weight, or, in other words, by increasing the diameter of the wheel, the weight may be reduced without loss of efficiency. In an automobile there is not often room for a

flywheel of large diameter, so that the rim must be made wide in order that the weight may be as far from the centre as possible, where its greatest effectiveness will be secured. As the duty of a flywheel is to act as a reservoir of energy, the effect of the other revolving masses, such as the clutch and gear wheels, the road (driving) wheels, and the weight of the vehicle itself, when all these are in motion, may be regarded as assisting the flywheel. When considering the speed variation of a motor, the difference in speed between no load and full load is not a matter for flywheel regulation, but for the governor. It is the steadiness in speed between the impulses that the flywheel is intended to effect, and in this matter the governor has no part. The degree of steadiness required for an automobile engine not being so great as for a stationary engine, a lighter flywheel can be employed. The following formulæ take into account the steadiness between the impulses, so that the designer can make his own choice. The permissible variation in speed can best be expressed as a coefficient, and the value of this for dynamo driving will be 0·01, but for a vehicle motor can be from 0·03 to 0·05. The following formulæ give the weight of the rim of the flywheel :—

$$(31) \qquad W = \frac{322000AV\left(\sqrt[3]{\frac{V}{V_1}} - 1\right)a}{D^2nN^2}$$

$$(32) \qquad W = \frac{322000AV\left(\sqrt[4]{\frac{P_1}{P}} - 1\right)a}{D^2nN^2}$$

where W = the weight of the flywheel rim in pounds, D = the mean diameter of the rim in feet, N = the number of revolutions per minute, a = the maximum number of idle strokes between the impulses, and n =

the coefficient of speed variation allowed. For a single-cylinder motor having one impulse stroke in every four, the value of a may be taken as 4, that is, three idle strokes plus one to allow for the work absorbed in compressing the charge. The symbols P, P_1, V, V_1 are as on p. 11. The value of n is given above. For safety the speed of the flywheel rim should not exceed 6000 feet per minute, or a maximum diameter of $D = \frac{1900}{N}$, where D = the diameter of the rim in feet, and N = the maximum number of revolutions per minute.

As an example we may take the motor considered on p. 13 in reference to the cylinder formulæ. As there are to be two cylinders, we may take the value of a as 2, and allowing a speed variation of 3 per cent., n will be = 0·03. Hence, substituting known values in formula 31, we have—

$$W = \frac{322000 \times 15{\cdot}9(0{\cdot}723)\left(\sqrt[2]{\tfrac{10}{3}} - 1\right)2}{1{\cdot}5^2 \times 900^2 \times 0{\cdot}03} = \frac{201{\cdot}53}{3{\cdot}04} = 66{\cdot}2 \text{ lbs.}$$

In this example the mean diameter of the rim has been assumed as 1·5 feet, and the value of V is also taken in feet, *i.e.* 8·68 inches = 0·723 foot. Having the *mean* diameter of the rim and the weight required, the width of rim necessary can be easily calculated.

Although, as stated above, the flywheel has no part in regulating the variation in speed between no load and full load, it may be taken that a heavy flywheel will materially assist the action of the governor. The inertia of the heavy wheel will tend to prevent sudden changes of angular velocity, and so give the governor time to act.

Balancing.—It is universally accepted that it is impossible to balance a reciprocating weight with one that is revolving, hence a single-cylinder engine cannot be

perfectly balanced. When more than one cylinder is employed, the reciprocating masses balance each other to a certain extent, as do also the swinging weights of the connecting rods. Motors with three or four cylinders can be made to run in almost perfect balance, so far as the moving parts are concerned. With a single-cylindered engine the writer is of the opinion that the weight of the piston can be entirely neglected, and only the crank pin, crank webs, and as much of the connecting rod as can be regarded as a rotating weight, need be provided for by balance weights. To determine how much of the total weight of the connecting rod to allow for, it should be

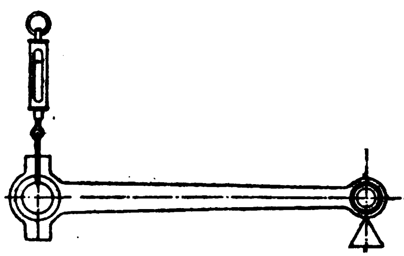


Fig. 13.

weighed in the following manner. Support the piston end of the rod on a knife edge at a point opposite the axis of the piston pin, and let the other end of the rod also rest on a knife edge which is carried on a scale or spring balance. The rod should be kept as nearly as possible in a horizontal position, and the weight as given by the spring balance will be the amount to allow for in the balance weight, as representing the revolving mass of the rod. This method of weighing the rod is illustrated in Fig. 13. A rough-and-ready approximation is to allow half the total weight of the rod as the rotating weight.

No general formula has as yet been evolved for

arriving at the correct weight of the balance weights, but the following will be found to give good average results. For special cases the weight as given by the formula may be taken as a basis for experiment.

Let B = the weight of all the balance weights.
M = the weight of the crank pin plus the rotating weight of the connecting rod.
J = the weight of the unbalanced portion of both crank webs.
m = the radius of the crank-pin path in inches.
j = the radius of centre of gravity of the crank webs in inches.
q = the radius of centre of gravity of the balance weights in inches.

The value of the factors B, M, and J are all to be taken in the same units, either ounces or pounds.

$$B = \frac{Mm + Jj}{q} \tag{33}$$

The force due to the inertia of the reciprocating parts acts along their line of motion, and will be at a maximum value at the commencement and end of each stroke. At about the middle of the stroke the value is zero. Neglecting the effect of the connecting rod, the maximum value of this force is found by the usual formula for centrifugal force—

$$F = 0{\cdot}00017N^2WS \tag{34}$$

in which F = the force in pounds, N = the number of revolutions per minute, W = the weight of the reciprocating parts in pounds, and S = the stroke in feet. To facilitate calculations, the values of the expression $0{\cdot}00017N^2$ for

various speeds have been calculated, and are tabulated below—

TABLE 3.

N	$0 \cdot 00017 N^2$	N	$0 \cdot 00017 N^2$
650	71·825	950	153·425
700	83·3	1000	170·0
750	95·625	1050	187·425
800	108·8	1100	205·7
850	122·825	1150	224·825
900	137·7	1200	244·8

The forces due to the acceleration of the reciprocating parts may be graphically represented by means of an ordinary diagram of forces, as in Fig. 14. In this diagram the vertical lines are divided to represent the force in pounds, and the horizontal line the stroke, both to any convenient scale. For the purpose of an example, the diagram has been drawn for a motor with a stroke of 5 inches, running at 600 revolutions per minute, the reciprocating parts being assumed as weighing 8 lbs. Substituting these values in formula 34, we have—

$$F = 0 \cdot 00017[(600)^2 8 \times 0 \cdot 417] = 202 \text{ lbs.}$$

Laying off this value upwards at one end of the stroke, and downwards at the other, and connecting these points, we get the line AB; the ordinates represent the forces. If the effect of the connecting rod is to be taken into account, we should increase the length of the vertical line, which represents the force at the commencement of the stroke, by the fraction of its length equal to the length of the crank divided by the length of the connecting rod, generally about 0·2. The other vertical line, representing the force at the end of the stroke, should be shortened by

an equal amount. Joining these points, we shall have the dotted line *ab*.

As the magnitude of the forces which require to be balanced depend on the weight of the parts in motion, these should be made as light as possible, consistent with the necessary strength.

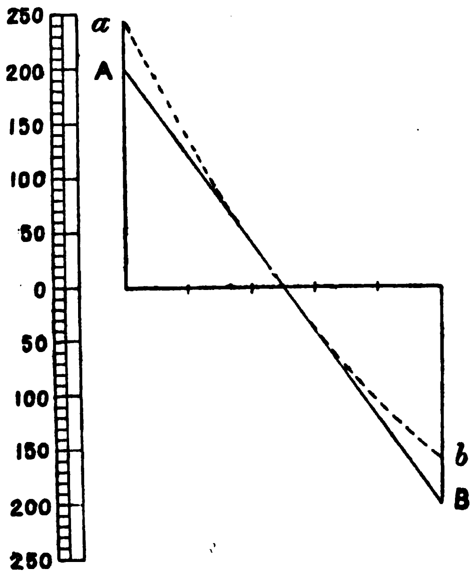


FIG. 14.

So far we have considered only the balancing of the moving parts of the motor. There still remains one force which cannot be balanced except by introducing an equal force acting in the opposite direction. The force referred to is the reaction due to the sudden combustion of the charge in the cylinder. This force can only be neutralized by exploding the charge between two oppositely moving pistons in one cylinder, or simultaneously in two cylinders

opposed to and in line with each other. The Gobron-Brillié and Koch motors are examples of the first, and the Lanchester and some American engines illustrate the second method. The Gobron-Brillié and the Koch engines both have a system of levers for transmitting the power from the two pistons to a common crank shaft, and these levers, with their connecting links, introduce vibrations of their own. In the usual type of opposed cylinder motors the two cylinders are slightly out of line, in order that a two-throw crank shaft may be used without having to make the connecting rods eccentric with the cylinders. From this a certain amount of wrenching ensues.

In the motor illustrated in Fig. 15 an attempt has been made to obtain perfect balance of the moving parts, and of the reaction due to

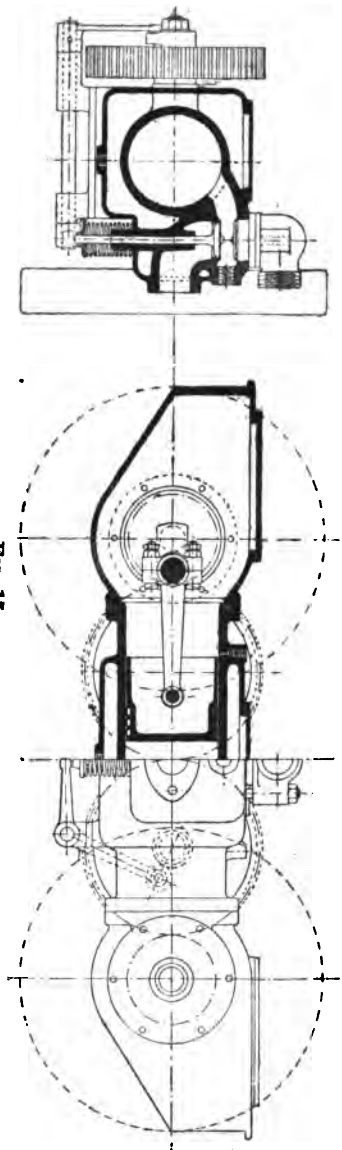


FIG. 15.

the explosion.[1] The writer had a considerable share in the design of this engine, and also in running it for testing purposes, and can testify to the total absence of vibration. The swinging levers and links which are found in the Gobron-Brillié and Koch motors are replaced by gear wheels, which serve to couple the two crank shafts. The two larger (intermediate) gear wheels are utilized, one to operate the exhaust valve, and the other the electric ignition cam, each of these wheels being half the size of those on the crank shafts. The crank shafts revolve in opposite directions. In the original design of the engine the two crank shafts both revolved in the same direction, only one intermediate wheel being used, as shown in Fig. 16. The balance in this form of engine was on the whole good, but not perfect, hence two intermediate wheels were used to cause the cranks to turn oppositely. A fault of the design is the great length of the engine, which makes it somewhat unsuitable for motor-car work. Possibly by employing two cylinders of short stroke a motor could be built on the same lines which would be compact enough and yet develop sufficient power.

Carburettors.—Although the petrol motor has been in every day use for a fairly long time, there is very little data on record dealing with explosive mixtures of petrol vapour and air. For want of more exact data, we must base our deductions, to a certain extent, on the behaviour of mixtures of coal-gas and air, and within limits the analogy will be close enough for all practical purposes.

Petrol varies in density (at 69° Fahr.) between 0·680 and 0·710 (76° to 68° Baumé). The boiling-points at these two densities are 149° Fahr. and 194° Fahr. respectively. The chief constituent of the vapour formed by the evaporation of petrol is pentane, having a specific gravity

[1] Patent No. 2847 of 1899. F. C. Nunn and T. H. White.

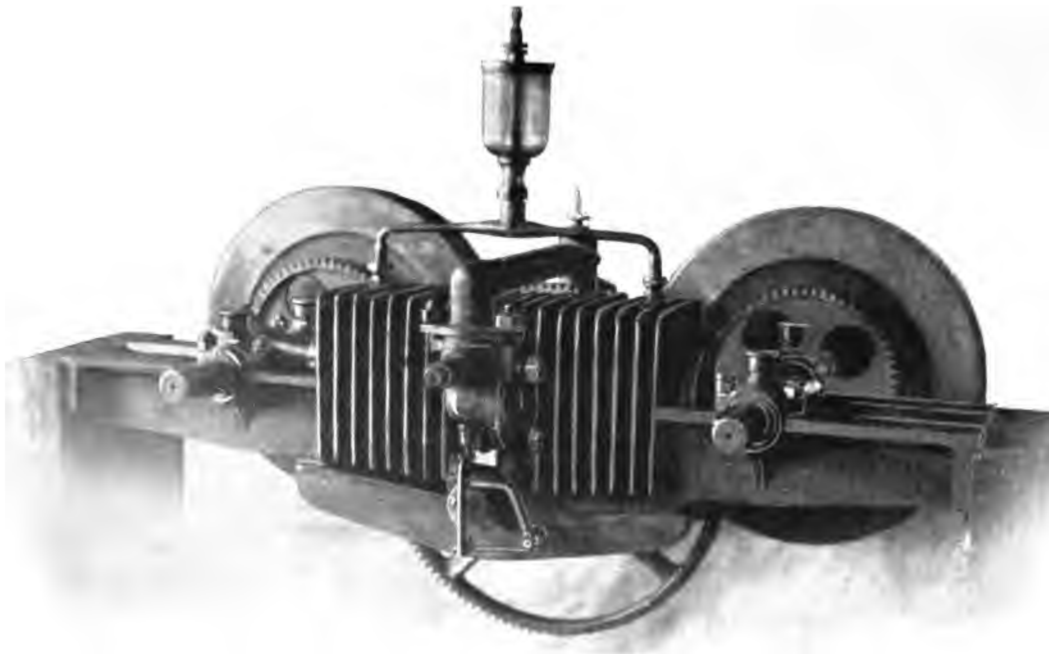


Fig. 16.

[To face p. 48.

of 0·640, and a chemical composition of C_5H_{12}, the liquid itself being a mixture of hexane and heptane, the proportions varying with the specific gravity; hexane (C_6H_{14}) having a specific gravity of 0·676, and heptane (C_7H_{16}) of 0·718.

The composition by weight of petrol having a specific gravity of 0·683 and a boiling-point of 154° Fahr. is—hexane, 80 per cent.; heptane, 18 per cent.; and pentane, 2 per cent. The chemical composition is, carbon 83·8 per cent. and hydrogen 16·2 per cent., corresponding to the formula $41 \cdot 86 C_6H_{14} + 6 \cdot 48 C_7H_{16} + C_5H_{12}$. It requires about 3·5 lbs. of air to consume 1 lb. of petrol, corresponding to a mixture of 100,000 volumes of air to 12·4 volumes of liquid petrol. The density of the vapour from petrol of the above specific gravity and chemical composition is about 3·05, and 1 kilogram of petrol vapour has a volume of 0·254 cubic metre. The proportions of petrol vapour and air, by volume, to give the greatest explosive effect, are therefore—

$$\frac{0 \cdot 254}{11800} = 2 \cdot 15 \text{ per cent.}$$

For convenience of reference, we may tabulate the various properties of petrol thus—

Specific gravity	0·680 to 0·710.
Boiling-point	149° to 194° Fahr.
Chief constituent	Hexane (C_6H_{14}).
Volume of air for perfect combustion, per kilo. .	11·8 cubic metres.
Proportion of liquid to air for perfect combustion	12·4 to 100,000.
Proportion of vapour to air, by volumes	2·15 per cent. vapour.

The following table shows the specific gravities corresponding to Baumé degrees at 60° Fahr. :—

TABLE 4.

Baumé degrees.	Specific gravity.	Baumé degrees.	Specific gravity.
63	0·728	72	0·695
64	0·724	73	0·692
65	0·720	74	0·689
66	0·717	75	0·685
67	0·713	76	0·682
68	0·709	77	0·679
69	0·706	78	0·675
70	0·702	79	0·672
71	0·699	80	0·669

For every eight degrees above 60° Fahr. one degree Baumé should be subtracted from the hydrometer reading, and for every eight degrees below 60° Fahr. one degree is to be added to the Baumé degrees. With a mixture of correct proportions the whole of the vapour will be consumed, and no objectionable odour or fouling of the engine will occur. The limits for the proportions of air and gas for complete combustion are fairly close, hence a carburettor requires to be carefully made and adjusted to ensure a constant supply of correctly proportioned mixture. Experience proves that a mixture of one volume of liquid petrol to about 8380 volumes of air gives good results. With these proportions combustion is rapid and the exhaust clean. Very little odour is to be noticed, and no fouling of the valves and ports. The proportions of the mixture will have to be varied somewhat, according to the quality of the spirit and the state of the atmosphere.

With more than 10,000 volumes of air to 1 of liquid petrol the mixture will not explode properly. Therefore it is advisable to keep the ratio of liquid petrol to air somewhere between 1 to 8000 and 1 to 10,000, which ratios correspond to about 1·9 per cent. and 2·4 per cent. of vapour in each case.

Mixtures are explosive up to a ratio of liquid petrol to air of about 1 to 4000, but when the ratio is reduced to 1 to 3400, the mixture will not be combustible. The ratio of liquid to air of 1 to 4000 is equivalent to about 4 per cent. of vapour, and 1 to 3400 to about 5·5 per cent. At 60° Fahr. air will not be saturated with petrol vapour till it has absorbed about 15 per cent. by volume.

At the present time one hears but little of the surface carburettor, except for motor cycles. The defects of this type were that the lighter constituents of the petrol were apt to be evaporated first, leaving a residue of greater density, and the petrol was splashed about too much. Both of these defects necessitated a constant alteration of the air-valve to maintain the mixture somewhere near its proper proportions. Surface carburettors are more economical of fuel than the jet type, owing probably to the more perfect mixture of the vapour with the air. Also the air usually has considerably more freedom allowed for its passage through a surface carburettor than one of the jet type; hence more power for a given-sized motor can be secured.

Surface carburettors should be so proportioned that the air will pass through them at a speed not exceeding 80 feet per second. In a jet-type carburettor a good suction effect is required, so that the speed of the air may be increased to 100 feet per second.

The suction effect at the jet varies as the *square* of the velocity of the air, so that when the motor increases its speed the proportions of the mixture are apt to be altered. Many devices are in use to prevent the mixture varying in quality, chiefly consisting of extra inlets for admitting air, operated by the suction effect of the piston, when the speed exceeds a predetermined limit. The writer proposes the form of carburettor shown in Fig. 17. The float chamber A is of the usual construction, and requires no

further description here. The jet B is surrounded by the inducing tube C, which is in one piece with the throttle valve D. This throttle valve is connected to the engine governor, so that as the speed of the motor falls the piece D will be raised. At the same time that the throttle valve is opened the effective area of the inducing tube will be

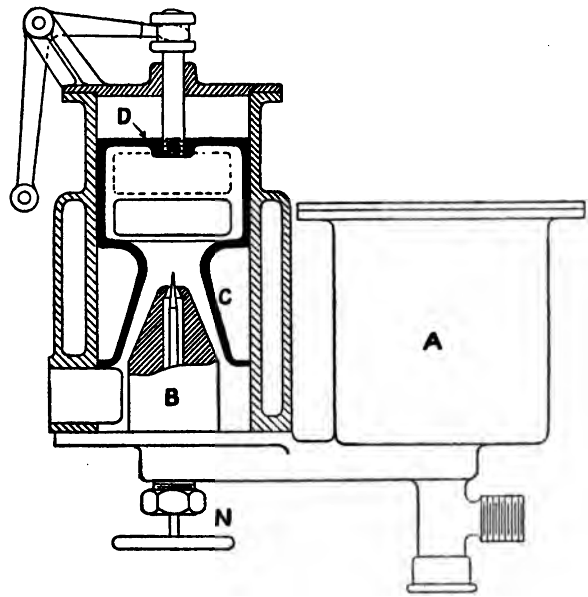


FIG. 17.

increased, owing to its form, and thus the air will not greatly increase its velocity. If carefully proportioned, a carburettor on these lines could be constructed to maintain the mixture constant in quality. To compensate for variation in the specific gravity of the petrol, or the state of the atmosphere, a needle valve is provided at N to control the size of the jet aperture. Other things being

equal, the efficiency of a carburettor is the measure of the freedom with which the mixture passes to the engine. Were it not for the unequal vapourization of the petrol, and the constant variation in the proportions of the mixture, surface carburettors would be preferable to the jet type if only on account of the free passage of the air and gas through them. One other defect of the surface type is the liability of explosion should a flame find its way into it, as even at temperatures considerably below 32° Fahr. there will be an explosive mixture formed.

Owing to the rapid evaporation of the petrol, heat is quickly absorbed from the metal forming the walls of the vapourizing chamber; hence provision must be made for supplying the heat necessary for the evaporation of the spirit. The usual plan is to provide the vapourizing chamber with a jacket, through which the heated water from the cylinder jackets, or a part of the exhaust gases, is allowed to circulate. The use of the water circulation is preferable to that of the exhaust gases, as the temperature is more likely to be kept even.

Governing.—There are two principal methods of governing the speed of a petrol motor. Either the force of each impulse may be varied, or the number of impulses in a given time may be changed. All motors that are provided with a governor are controlled by one of these two methods, or some modification thereof. By the first method the force of the explosion is diminished by admitting a smaller charge into the cylinder. The efficiency of an engine governed on this system will not be so high as when the governing is effected by cutting out the impulses entirely, when it becomes necessary to reduce the speed. In the first method, by reducing the quantity of air and gas taken into the cylinder, the compression pressure is lowered, and this does not tend to make the engine economical, as explained on p. 8. But the turning

moment of the crank shaft is more even than when the charges are cut out altogether. The diminution of the amount of the charge can be effected in a variety of ways. A throttle valve in the pipe leading from the carburettor is the most usual device, but the same end may be accomplished by altering the lift of the inlet valve, or the time it remains open. The Crossley motor employs an auxiliary cut-off valve, through which the mixture has to pass on its way to the inlet valves, and which is acted upon by the governor to cut off the supply of air and gas before the suction stroke is completed, when the speed of the engine increases to such a point as to render this desirable.

Governing by cutting out the impulses may be effected in two ways when automatic inlet valves are used—either by retaining the products of combustion in the cylinder by causing the exhaust valve to remain inoperative during one or more cycles, or by allowing the exhaust valve to remain open during one or more suction strokes, either way preventing the formation of a sufficient vacuum to open the inlet valve. The first of these methods, *i.e.* keeping the burnt gases in the cylinder, was the system adopted in the original Daimler motors, and was economical of fuel. By suitably designing the cams which operate mechanically opened inlet valves, the governor may be made to render these cams inoperative for as long as may be required to bring the engine speed down to the normal, or the valve rods may be acted upon with similar effect.

It is possible to vary the speed of the engine by altering the timing of the ignition; but this is not to be recommended, as it is wasteful of fuel. By governing in this manner, the amount of fuel consumed will remain constant at all loads. The only use for this way of altering the engine speed is for temporary occasions, when other means are not so convenient, or in emergencies.

For stationary motors, such as those used for dynamo driving, the best system of governing is that in which the impulses are cut out entirely, as the utmost fuel economy is obtained, and the slight irregularities in speed can be compensated for by having a heavy flywheel. This system would also find acceptance for marine work. In the case of an automobile it is necessary to have some means of regulating the speed of the motor, within fairly wide limits, from the driver's seat, and in this connection there is nothing better than the throttle valve. The hand-operated throttle valve gives the driver the power of adjusting the speed of the vehicle without constant recourse to the speed gearing, or rather it gives a means of control supplemental to that of the gearing.

Upon whatever system the governor works, it should be so designed that the driver of the car can nullify its action at will, when the greatest speed is required from the motor. It is good practice to arrange for an auxiliary throttle valve connected to the brake gear, so that when the vehicle is stopped the motor will be automatically slowed down, thus avoiding waste of fuel, and preventing undue vibration.

The actual design of the governor may be left to individual judgment. Formulæ are of little value in connection with such small governors as are required in automobile work, and, moreover, there is generally a good deal of latitude allowed in the adjustment of the springs. The governor weights need only be small, even for high-powered engines, as the work imposed upon the governor is usually very slight.

Ignition.—At the present time the ignition of the charge by an electric spark may be said to be universal, the hot-tube method having been quite abandoned for automobile engines. The only advantage to be claimed for the hot-tube method of ignition is its reliability and

simplicity. It is, however, not adjustable in regard to timing the moment of igniting the charge, and thus motors equipped with it can only be run at one speed economically. It is obvious that there is a risk of fire should anything cause an upset of the petrol, and this was by no means an unknown danger in the days when tube ignition was universal, or practically so. For stationary work, such as pumping or dynamo driving, tube ignition still has its uses, and is in many cases to be preferred to electric ignition for such purposes.

There are two systems of electric ignition in use, either of which is capable of giving satisfactory results, provided it be properly installed and maintained in working order. These two systems are known as the low-tension and the high-tension. The low-tension system requires a make-and-break device inside the combustion chamber, operated from the outside, usually by a cam on the valve-gear shaft. The system has the advantage that all the wires are easily insulated, owing to the low voltage of the circuit. The necessary fittings, with the exception of the sparking device in the combustion chamber, are cheap, and require few repairs.

In the high-tension system there are no moving parts within the cylinder, but there is greater difficulty in insulating the conductors of the secondary circuit, owing to the high tension, which may be as great as 30,000 volts. In the matter of cost of installing, it is believed that there is very little to choose between the low and high tension systems. An efficient low-tension coil may be constructed as follows. The core should consist of a bundle of soft iron wire of about 20 B.W.G., and should be about 9 inches long by 1 inch diameter. A thin tube of insulating material is placed over the core, and upon it is wound double cotton-covered wire of 14 B.W.G. till there are three layers. The connections for such a coil

are seen in Fig. 18, in which A is the coil, B the source of current, and C the make-and-break device in the combustion chamber. With this system it is most important that the break between the sparking points in the cylinder should be as rapid as possible, and it is this particular which forms the principal claim in the numerous patents on the subject. The sparking points require to be in

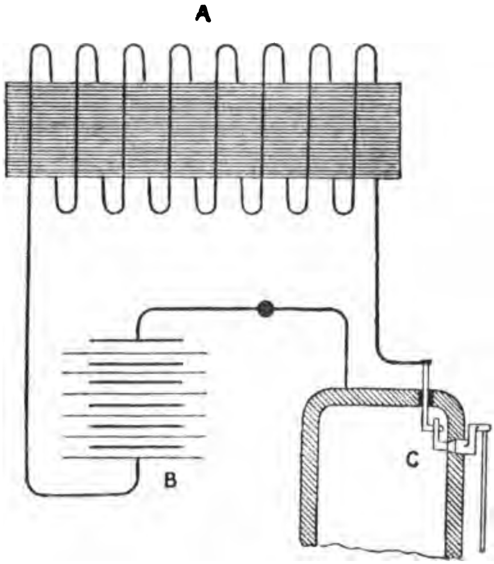


FIG. 18.

contact long enough to ensure thorough energizing of the magnet, and with fast-running motors this matter becomes important. Too short a contact will so reduce the spark as to render the ignition of the charge very uncertain, and too long a contact will be wasteful of current. The same remarks apply to high-tension coils. The current should be allowed to flow, in either high or low-tension coils, for from 0·03 to 0·05 second.

As the "sparking length" of a coil is considerably reduced when the discharge takes place in a dense medium, such as the compressed charge in the combustion chamber, it is advisable to have a coil capable of giving a spark quite $\frac{5}{8}$ inch long in air. As this will require an electro-motive force of fully 30,000 volts, and possibly more, the necessity for perfect insulation will be evident.

The use of an external spark-gap in series with the sparking plug is not always to be advised. The increased resistance offered to the passage of the secondary current increases the risk of the discharge taking place inside the coil itself, and once this occurs, the coil will be ruined. In any case, the external spark-gap strains the insulation of the secondary winding, so to speak, and when it is intended to use this accessory the coil should be specially insulated.

For a coil to give a half to one inch spark in air the following notes and dimensions will give satisfactory results. The core, of a bundle of well-annealed soft iron wire, should be 7 inches long by $\frac{3}{4}$ inch diameter. The wire must be in perfectly straight pieces, and No. 22 B.W.G. in thickness. The core is insulated with linen tape, wound on spirally in three layers, each layer well soaked in shellac varnish. The primary coil is wound directly on the insulated core, and should consist of two layers of No. 18 B.W.G. double cotton-covered wire, the length of this coil being about 6 inches, about half a pound of wire being required.

The insulation between the primary and secondary windings is of vital importance, and should take the form of a vulcanite, or fibre, tube, $\frac{1}{8}$ inch thick in the walls, and be a good close fit over the primary winding. At each end of the tube, hard wood, or vulcanite, cheeks are to be fitted to form a bobbin, upon which the secondary is wound. The quantity of wire to be employed for the

secondary winding will depend on the length of spark desired. For a ½-inch spark, use half a pound; for a ¾-inch spark, three-quarters of a pound; and for a 1-inch spark, one pound. The gauge will be the same in each case, *i.e.* No. 36 B.W.G. double silk-covered. The wire must be free from kinks, and be tested from time to time to make sure that it is continuous. When fully wound, the whole coil should be soaked in hot paraffin wax, to exclude air and damp, and improve the insulation.

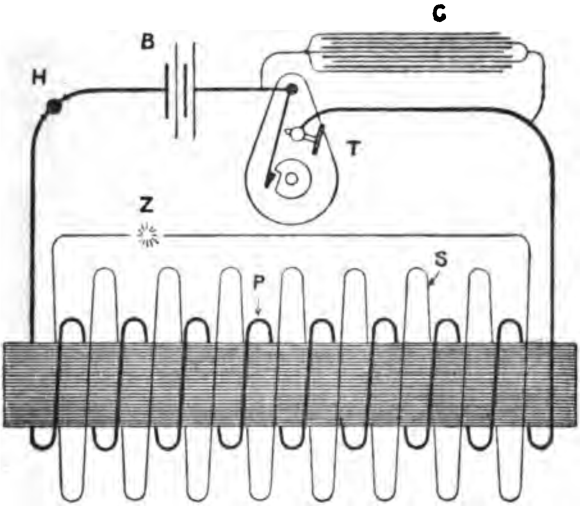


FIG. 19.

An efficient condenser is required, and for the three sizes of coils mentioned above this may consist of fifty, seventy, or ninety pieces of tin foil, each measuring 7 inches by 4 inches. The condenser is to be connected in shunt across the primary contact-breaker terminals as in Figs. 19 and 20. The details of coil construction will be found in more than one work dealing only with this matter, and need not be particularized here.

To a motor-car designer the arrangement of the circuits will be of more use than a description of coil-making. High-tension coils are made in two forms, *i.e.* with, and without, trembler. For a non-trembler coil the connections are as seen in Fig. 19, and for a coil having a trembler as an integral part of its construction the connections will be found in Fig. 20. In each of these figures B is the source of current, P is the primary winding, S the secondary winding, T the trembler or contact breaker, C the con-

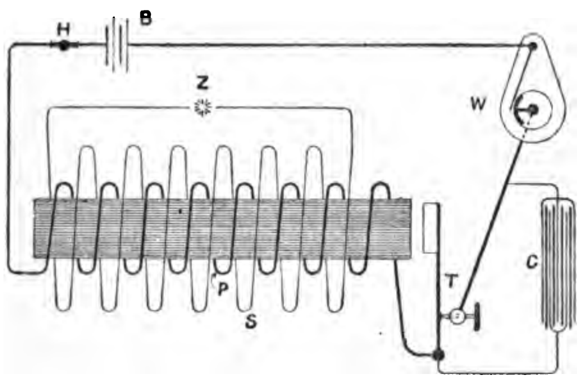


FIG 20.

denser, H a switch for breaking the primary circuit when desired, and Z the sparking points to be located within the combustion chamber. With a non-trembler coil there is no need for an additional make-and-break device operated by the engine, as the trembler itself does this duty; but when the trembler forms part of the coil, a contact mechanism, often miscalled the commutator, is required. In Fig. 20 this is indicated by W, and usually consists of a simple wipe contact. The rotating contact strip requires to be designed with a view to the current being allowed to flow for a sufficient time for energizing the coil at the highest speed

it is intended to run the engine. Obviously this will involve a certain waste of current at slow speeds, but to compensate the apparatus to give equal time of contact at all speeds would probably add more complication than the saving in current consumption would warrant. Also, it should be remembered that the motor will generally be worked at a speed more nearly approaching the maximum than the minimum. The timing of the moment of ignition is usually regulated by hand, but it would seem advisable to provide automatic means for doing this. With a motor of varying speed it is a matter of impossibility for any one to so adjust the contact device as to ensure the charge being ignited at the proper instant at all speeds. No harm can be done to the engine by setting the apparatus to fire the charge late, but if the spark is produced too early, by advancing the lead of the contact piece too much, the motor may be wrecked. There are plenty of cases on record where the connecting rod has been doubled up and the crank shaft bent or broken by giving the ignition too much lead. Wherefore some kind of centrifugal governor is desirable whereby the ignition shall always take place at the proper point in the cycle. In addition to reducing the number of levers requiring the attention of the driver, such a device would be a safeguard against premature ignition when the motor is started.

The writer has used an ignition-timing governor of the type shown in Fig. 21. The insulating disc carrying the contact strip is fastened upon the boss B, the position of which in relation to the gear wheel is controlled by the weighted arms A and A′. As these arms move outwardly towards their extreme position, which is shown in dotted lines, the disc is given more or less angular advance, thereby advancing the moment of ignition as the speed of the engine increases. As the speed drops, the arms will resume their normal position under the influence of the

springs, and the moment of ignition will be retarded. When the motor is being started from rest the ignition gear will be in its most retarded position, thus avoiding all risk of back-firing. The governor is adapted from the crank-shaft expansion governor fitted to steam engines. The important point is to adjust the spring tension correctly.

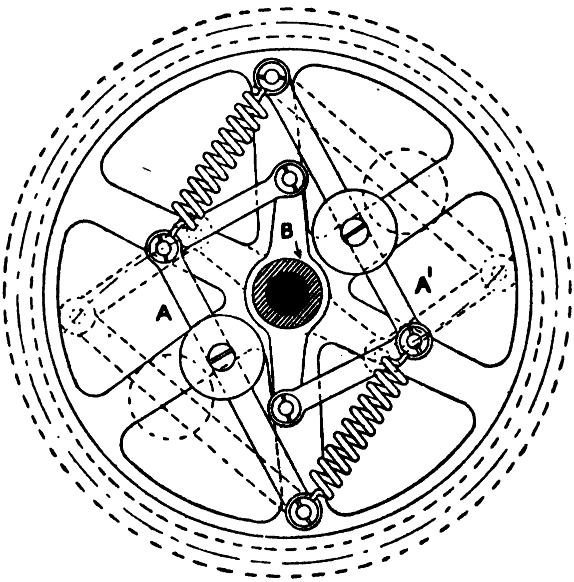


FIG. 21.

To supply the current necessary for the production of the spark, primary or secondary batteries, magneto or dynamo electric machines, are used. Magneto machines are used without a coil if the low-tension system is employed with a make-and-break inside the cylinder, but they may be used in conjunction with an induction coil and the usual sparking plug. Dynamos are not much used in this country, but in America there are several

makes of cars which have both dynamo and storage batteries, which can be used alternately at the will of the driver. With this arrangement the secondary battery is always kept fully charged; an automatic cut-in and cut-out is fitted so that the dynamo is only in circuit when running at its proper speed, when the battery is cut out. This system would seem to promise well, but the dynamo requires designing so that its output is fairly constant at varying speeds.

General Design.—The relative advantages of horizontal and vertical engines have been the subject of much discussion in the past, and the vertical engine has so far been more generally employed, in this country and in France and Germany. In America the preference seems to be for horizontal motors. Both designs have their good and bad points fairly evenly balanced, but the writer inclines to the horizontal engine. With a vertical engine the vibration is more evident. The direction of movement of the disturbing forces in a horizontal motor is all in line with the axis of the car, in which direction they can best be resisted, whereas with a vertical motor the disturbing forces have only the springs to resist them. It has been advanced that the cylinder of a horizontal motor will wear oval in a much shorter time than when vertical, the contention being that this wear is caused by the weight of the piston. It is very much to be doubted whether the weight of the piston has any influence on the wear. The chief factor is the pressure due to the angular thrust of the connecting rod, and this will be practically the same in both vertical and horizontal cylinders. On the question of lubrication of the piston, it would seem reasonable to suppose that when this is effected by "splash" only, the vertical position will be best, but when the oil is introduced through the side of the cylinder, the horizontal position will secure better distribution.

To avoid forming shoulders in the cylinder bore by the wear of the piston, it is usual to allow the piston to move a short distance beyond the actual bored length by enlarging the diameter of the cylinder at the combustion chamber, and bell-mouthing the open end. The change in diameter at the combustion chamber should not be

TABLE 5.

TABLE OF STUD DIMENSIONS.

Diameter of stud. Inches.	Dimensions in inches.									Threads per inch.
	A	B	C	J	E	F	G	H	I	
1/8	1/4	3/16	3/8	3/8	3/16	1/4	3/8	25/64	11/32	40
3/16	5/16	1/4	1/2	9/16	1/4	3/8	7/16	33/64	7/16	24
1/4	3/8	5/16	5/8	5/8	5/16	7/16	9/16	39/64	33/64	20
5/16	1/2	3/8	3/4	3/4	3/8	1/2	5/8	11/16	39/64	18
3/8	5/8	7/16	7/8	15/16	7/16	9/16	3/4	13/16	45/64	16
7/16	3/4	1/2	1	1 1/16	1/2	5/8	7/8	61/64	55/64	14
1/2	7/8	9/16	1 1/4	1 1/4	5/8	7/8	1	1 1/16	29/32	12
5/8	1	3/4	1 7/16	1 1/2	3/4	1	1 1/4	1 17/64	1 3/32	11
3/4	1 1/8	7/8	1 5/8	1 11/16	3/4	1	1 1/2	1 1/2	1 19/64	10
7/8	1 1/4	1	2	2	15/16	1 3/16	1 5/8	1 45/64	1 31/64	9
1	1 3/8	1 3/16	2 3/16	2 3/16	1	1 1/4	1 7/8	1 61/64	1 43/64	8
1 1/8	1 1/2	1 5/16	2 7/16	2 1/2	1 1/8	1 3/8	2	2 5/32	1 57/64	7
1 1/4	1 5/8	1 7/16	2 11/16	2 3/4	1 1/4	1 5/8	2 1/4	2 23/64	2 3/64	7
1 3/8	1 3/4	1 9/16	2 15/16	3	1 7/16	1 7/8	2 1/2	2 35/64	2 7/32	6
1 1/2	1 7/8	1 3/4	3 1/16	3 1/4	1 5/8	2 1/8	2 3/4	2 25/32	2 13/32	6

abruptly made by stepping, but the surface should be tapered from one diameter to the other. If made by an abrupt step, and the piston should be pushed too far up the bore, one, or more, of the piston rings will spring out into the combustion chamber, and will prevent the piston being removed without breaking either it or the ring. If

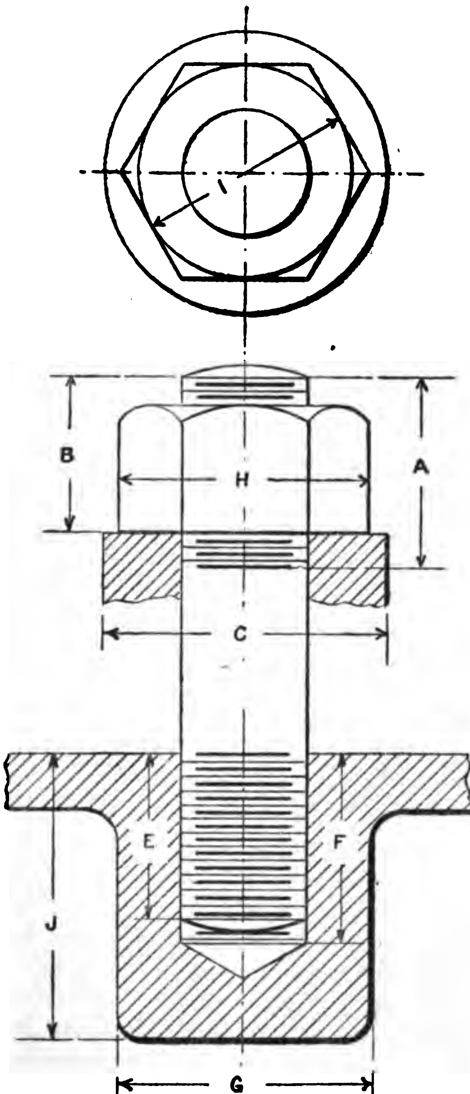


FIG. 22.

the bore is tapered, however, the piston can be withdrawn without much difficulty, as the taper will act to close the ring back into its groove.

With any piece of machinery it looks bad to see the nuts overhanging the facings upon which they bear, or to see too great a surface of the facing showing round the nut. For some time the writer has used the dimensions in Table 5 where studs or bolts have been required in a design, and has found a considerable saving in time thereby. The dimensions given for the boss into which the stud is screwed apply more particularly to cases where it is not advisable for the end of the stud or screw to come right through, such as a cylinder water-jacket.

In all cases where two parts of an engine are bolted together, and where the edges of the parts are not machined, such as the cylinder and crank chamber, the upper piece should be slightly smaller than the lower, to give a little freedom in placing the parts while avoiding overhang. This applies specially to pipe flanges; there should always be from a sixteenth to an eighth of an inch of the facing showing all round the edge of the flange, unless the edges of both flange and facing are machined flush with each other.

The two following tables of flange dimensions will be useful in designing motors, Fig. 23 being for ordinary wrought-iron gas-pipe sizes, the pipe being screwed into the flange; and Fig. 24 for brass or copper tube, brazed into the flange. The dimensions in Fig. 24 might also be adopted where weldless steel tube of thin gauge, say up to No. 16 B.W.G., is in question; for thicker gauge tubing use Fig. 23. Where it is necessary to employ coned unions, the dimensions in Fig. 25 and Table 8 can be followed. The ordinary coned unions used for connecting up gas-piping will generally require regrinding, with a very fine

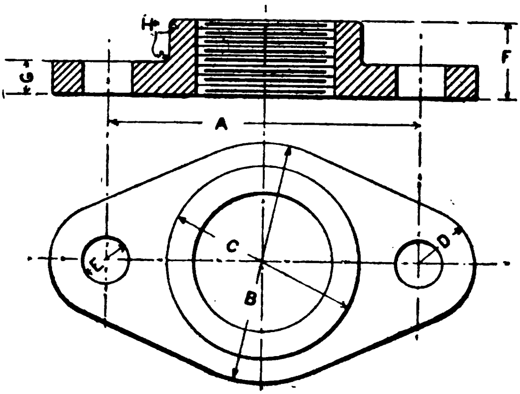


FIG. 23.

TABLE 6.

CAST-IRON PIPE FLANGES.

(Dimensions in inches.)

Size of pipe.	A	B	C	D	E	F	G	H	Pipe diameter.	Tapping diameter.	No. of threads.
$\frac{1}{8}$	$1\frac{5}{8}$	$1\frac{3}{16}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{16}$	0·3825	0·3367	28
$\frac{1}{4}$	$1\frac{13}{16}$	$1\frac{3}{8}$	1	$\frac{3}{8}$	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{5}{16}$	$\frac{1}{16}$	0·518	0·4506	19
$\frac{3}{8}$	$2\frac{1}{8}$	$1\frac{5}{8}$	$1\frac{3}{16}$	$\frac{7}{16}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{1}{16}$	0·6563	0·589	19
$\frac{1}{2}$	$2\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{8}$	$\frac{3}{32}$	0·8257	0·7342	14
$\frac{3}{4}$	$2\frac{7}{8}$	$2\frac{3}{16}$	$1\frac{9}{16}$	$\frac{9}{16}$	$\frac{1}{2}$	$\frac{11}{16}$	$\frac{1}{2}$	$\frac{3}{32}$	1·04	0·9495	14
1	$3\frac{1}{4}$	$2\frac{1}{2}$	$2\frac{1}{16}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{3}{4}$	$\frac{9}{16}$	$\frac{3}{32}$	1·309	1·1925	11
$1\frac{1}{4}$	$3\frac{3}{4}$	$2\frac{7}{8}$	$2\frac{7}{16}$	$\frac{5}{8}$	$\frac{9}{16}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{3}{32}$	1·65	1·534	11
$1\frac{1}{2}$	$4\frac{1}{4}$	$3\frac{3}{16}$	$2\frac{13}{16}$	$\frac{11}{16}$	$\frac{5}{8}$	1	$\frac{5}{8}$	$\frac{3}{32}$	1·883	1·766	11
$1\frac{3}{4}$	$4\frac{13}{16}$	$3\frac{9}{16}$	$3\frac{3}{16}$	$\frac{11}{16}$	$\frac{5}{8}$	1	$\frac{11}{16}$	$\frac{3}{32}$	2·047	1·93	11
2	$5\frac{3}{8}$	$3\frac{7}{8}$	$3\frac{5}{8}$	$\frac{7}{8}$	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{3}{32}$	2·347	2·23	11
$2\frac{1}{4}$	6	$4\frac{1}{4}$	$3\frac{15}{16}$	$\frac{7}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{1}{8}$	2·588	2·47	11
$2\frac{1}{2}$	$6\frac{5}{8}$	$4\frac{13}{16}$	$4\frac{5}{8}$	1	$\frac{7}{8}$	$1\frac{5}{16}$	$\frac{7}{8}$	$\frac{1}{8}$	3·001	2·885	11
$2\frac{3}{4}$	$7\frac{5}{16}$	$5\frac{3}{8}$	$4\frac{15}{16}$	$1\frac{1}{16}$	1	$1\frac{7}{16}$	1	$\frac{1}{8}$	3·25	3·13	11
3	$7\frac{7}{8}$	$6\frac{5}{16}$	$5\frac{11}{16}$	$1\frac{3}{16}$	$1\frac{1}{8}$	$1\frac{9}{16}$	$1\frac{1}{8}$	$\frac{3}{16}$	3·485	3·368	11

grinding medium, to make them petrol-proof; as purchased, they nearly always leak. The small cocks required in

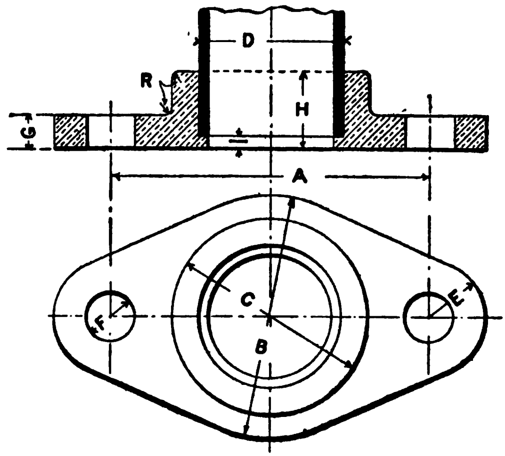


FIG. 24.

TABLE 7.

BRASS PIPE FLANGES.

(Dimensions in inches.)

Diameter of pipe. D	A	B	C	E	F	H	G	R	I	Bore of pipe.
$\frac{5}{8}$	$1\frac{5}{8}$	$1\frac{3}{16}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{2}$
$\frac{3}{4}$	$1\frac{13}{16}$	$1\frac{3}{8}$	1	$\frac{7}{16}$	$\frac{5}{16}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{5}{8}$
$\frac{7}{8}$	$2\frac{1}{8}$	$1\frac{5}{8}$	$1\frac{3}{16}$	$\frac{1}{2}$	$\frac{3}{8}$	$\frac{9}{16}$	$\frac{5}{16}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{3}{4}$
1	$2\frac{1}{2}$	$1\frac{7}{8}$	$1\frac{3}{8}$	$\frac{1}{2}$	$\frac{7}{16}$	$\frac{5}{8}$	$\frac{5}{16}$	$\frac{1}{16}$	$\frac{1}{8}$	$\frac{7}{8}$
$1\frac{1}{8}$ and $1\frac{1}{4}$	$2\frac{7}{8}$	$2\frac{3}{16}$	$1\frac{9}{16}$	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{3}{4}$	$\frac{3}{8}$	$\frac{1}{16}$	$\frac{1}{8}$	1
$1\frac{3}{8}$ and $1\frac{1}{2}$	$3\frac{1}{4}$	$2\frac{1}{2}$	2	$\frac{5}{8}$	$\frac{1}{2}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{16}$	$\frac{1}{8}$	$1\frac{3}{16}$

the petrol-pipe line will, unless specially made, also require regrinding; the best plan is to discard the use of

plug cocks altogether and use screw-down needle valves instead. As well as being proof against leakage, needle valves are less likely to become choked, being to a certain extent self-clearing. For making joints in the petrol pipes, red lead, rubber, and such-like are useless. Flexible vulcanized fibre can be relied on, and when something to take the place of red-lead cement is wanted, use soap. This, being insoluble in petrol, makes an excellent jointing material.

When making standard drawings for motors, it will be found convenient to keep cast and wrought work on separate sheets as far as possible, grouping the component parts with a view to the various shops concerned in carrying out the designs. The writer has also found it advisable to have two sizes of drawings—for general arrangements, 30 × 22 inches; and for details, 22 × 15 inches. The smaller sheets are more convenient for the workmen to handle, while the larger, not being in such constant use, do not get in the way.

Cooling.—Owing to the great heat developed within the cylinder of an internal combustion engine (see p. 19), it is necessary to employ extraneous means for keeping the cylinder cool enough to permit of proper lubrication. The cooling system is not intended, as is sometimes thought, to abstract the heat from the gases within the cylinder, but is provided to cool the cylinder walls only. The gases themselves should retain as much as possible of the heat due to combustion, hence it is advisable to let the cylinder work at as high a temperature as is consistent with efficient lubrication. It should also be the aim of the designer to arrange the cooling system so that the temperature of the cylinder may be kept as uniform as possible.

Air-cooling is limited to engines of small dimensions, though many attempts are being, and have been, made to

apply it to motors of forty horse-power, especially in America. Air-cooled engines are very liable to become overheated, when the piston is apt to bind in the cylinder. Also the incoming charge of air and gas is likely to be

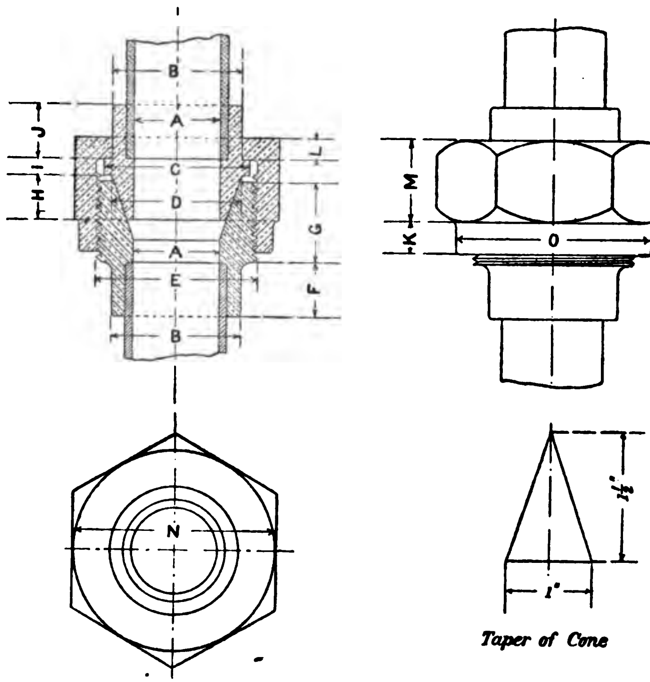


FIG. 25.

considerably attenuated by the expansion due to the mixture being heated as it flows into the combustion chamber. By the judicious use of fans, giving a rapid circulation of air around the combustion chamber, the effectiveness of air-cooling may be somewhat increased, but the fact that a fan takes a certain amount of power to

TABLE 8.

PIPE COUPLINGS.

(Dimensions in inches.)

Bore of pipe.	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2
A	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	2
B	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{5}{16}$	$2\frac{7}{16}$	$2\frac{9}{16}$	$2\frac{11}{16}$
C	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{5}{16}$	$1\frac{7}{16}$	$1\frac{9}{16}$	$1\frac{11}{16}$	$1\frac{15}{16}$	$2\frac{3}{16}$	$2\frac{5}{16}$	$2\frac{9}{16}$	$2\frac{11}{16}$	$2\frac{13}{16}$	$2\frac{15}{16}$	$3\frac{1}{16}$
D	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{8}$	$1\frac{3}{8}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{7}{8}$	2	$2\frac{1}{8}$	$2\frac{5}{16}$	$2\frac{7}{16}$	$2\frac{9}{16}$	$2\frac{11}{16}$
E	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	$1\frac{7}{8}$	$2\frac{1}{8}$	$2\frac{3}{8}$	$2\frac{1}{2}$	$2\frac{3}{4}$	$2\frac{7}{8}$	3	$3\frac{1}{8}$	$3\frac{1}{4}$
F	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$
G	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{15}{16}$	1	1	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{1}{16}$	$1\frac{1}{8}$	$1\frac{1}{8}$
H	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{13}{16}$	$\frac{13}{16}$
I	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$
J	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{11}{16}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$	$\frac{7}{8}$
K	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$
L	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$
M	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$	$\frac{15}{16}$	$\frac{15}{16}$	$\frac{15}{16}$	1	1	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{1}{4}$	$1\frac{3}{8}$	$1\frac{3}{8}$
N	$1\frac{1}{4}$	$1\frac{7}{16}$	$1\frac{5}{8}$	$1\frac{13}{16}$	2	$2\frac{3}{16}$	$2\frac{3}{8}$	$2\frac{9}{16}$	$2\frac{3}{4}$	$2\frac{15}{16}$	$3\frac{1}{8}$	$3\frac{5}{16}$	$3\frac{1}{2}$	$3\frac{11}{16}$	$3\frac{7}{8}$
O	$1\frac{3}{16}$	$1\frac{3}{8}$	$1\frac{9}{16}$	$1\frac{3}{4}$	$1\frac{15}{16}$	$2\frac{1}{8}$	$2\frac{5}{16}$	$2\frac{1}{2}$	$2\frac{11}{16}$	$2\frac{7}{8}$	$3\frac{1}{16}$	$3\frac{1}{4}$	$3\frac{7}{16}$	$3\frac{5}{8}$	$3\frac{13}{16}$
Threads per inch	12	12	12	12	12	12	12	8	8	8	8	8	8	8	8

drive it, especially at high speed, should not be lost sight of. Indeed, quite a large percentage of the extra efficiency due to the use of the fan may be discounted by the power required to drive it.

A properly designed water-cooling system will allow the engine to be worked at its maximum output, both for speed and power, for long runs, which is not possible with air-cooling. The honeycomb radiator, which for some time was practically universally adopted, is rapidly going out of fashion. The chief recommendation for its use was that the evaporation of the water was reduced to a minimum, and it made it possible to run a car for a week without replenishing the water-supply. The fact that it is possible to keep the water too cool was apparently lost sight of. By maintaining the temperature of the water at a point below that at which vapour is formed, the cylinders are unduly cooled, and a large proportion of the heat generated by the combustion of the charge goes to reheat the cylinder walls. In addition to this loss, there is the power required to drive the fan, which is necessary to cause the air to pass through the radiator, and the increased resistance offered to the circulation of the water through the restricted passages within the radiator. The Gillet-Forest system is unique, as the jacket water is allowed to boil and the radiator is utilized to condense the steam formed. The cylinder jacket is kept filled by a float valve, which allows water to enter to make up for that turned into steam. The engine is, with this system, worked at as high a temperature as possible, and the efficiency is very high.

Plain gilled copper tube of ⅝-inch or ¾-inch bore makes a simple and reliable radiator, but the gills should be of copper, and soldered to the tube. To secure a maximum cooling effect, the whole radiator should be finished dead black. The length of tube recommended is, for ⅝-inch

tube, 9 feet per *indicated* horse-power, and for $\frac{3}{4}$-inch tube, 6 feet per I.H.P. The diameter of the gills should not be less than twice that of the tube, and are best spaced about half the tube diameter apart. It is advisable to employ a pump to circulate the water through the cylinder jacket and radiator, as natural circulation cannot always be relied upon. The height of the column of water usually possible in an automobile is too little to cause a flow. The pump used should be of a type which will permit free flow of the water through it in the event of its ceasing to act.

It is important to so design the circulating system that no air-locks are formed, as this would interfere with the flow; and in any case it will be advisable to provide an air-cock at the highest point of the system, which can be opened when the engine is started till water shows, and thus prove that the pump is working. A drain-cock at the lowest point of the water system is a necessity, to enable the water to be run out in frosty weather, or when repairs are required. Neglect to empty the cylinder jacket and pipes has often resulted in a cracked jacket when the water contained has frozen. By dissolving chloride of lime in the cooling water, the temperature at which it will freeze is much below 32° Fahr., but the lime is apt to be deposited in the pipes when the water is heated, and on the whole its use is not to be recommended; the drain-cock is preferable. Heavy mineral oil has been tried as a substitute for water, but without much success. The difficulty appears to be in cooling the oil when it has been heated by the motor, as it does not part with its heat so readily as water, nor does it abstract the heat from the cylinder walls so quickly. The amount of water that should be carried on a car ought not to be less than half a gallon per indicated horse-power, and more if possible. By having a good body of water, the temperature is kept

more even. The temperature of the water as it leaves the cylinder jacket ought to be about 170° Fahr. If more, it indicates that too much heat is being abstracted from the engine, and the increased evaporation will cause the water to be used up too soon.

The great diversity of opinion among designers as to the dimensions of cylinders, valves, piston speed, etc., is forcibly illustrated by the annexed table of data, which formed part of a paper on valve gears read by Mr. Robert E. Phillips at the Automobile Club in 1904, and which is reproduced by its author's permission. From inspection of the table, it would appear that the rated powers of the motors cannot in all cases be true, and the need for some uniformity in design is apparent. In some examples the piston speeds are remarkably high, and the result would hardly tend towards economy in fuel consumption and upkeep.

In this table the diameters of the valves given are the diameters at the inner edge of the seating, and the mean and maximum velocities of both the charge and of the exhaust through the pipes are only given in those cases where the area of the pipe is less than that of the passage by the valve at its maximum lift.

It will be seen that the mean charge velocity through the most restricted area ranges from 58 feet to 448 feet per second. The mean exhaust velocity through the most restricted area varies from 52 feet to 346 feet per second.

Thirty-six of the engines are fitted with atmospheric induction valves, and fifty-seven with mechanically operated ones.

It will be seen to what extent some engines are throttled by reason of the small pipes employed. One of the worst offenders in this respect has a mean velocity of the charge through the inlet port of 141 feet per second, and through the induction pipe 376 feet per second.

PART II.

TRANSMISSION GEARING.

As the petrol motor is not self-starting, it is necessary to provide means for disconnecting the motor from the transmission gearing when the car is stopped temporarily, to avoid the restarting of the motor which would otherwise be required. In the case of gear-driven vehicles this is usually accomplished by means of a friction clutch. Those cars which have epicyclic gearing can be put out of gear by slackening the brake band on the slow-speed gear, but a friction clutch will generally be found to form part of the high-speed mechanism. Hence the design of friction clutches is an important part of the motor-draughtsman's work. Examination of a large number of cars has resulted in disclosing a great want of uniformity in the dimensions of their clutches, and this can only be explained by assuming that some are too small and some too large for the work imposed upon them. If too small, great pressure must be used to make the clutch transmit the power without slipping; if too large, it only means that an unnecessary amount of material has been employed; but if anything the clutch is all the better for it, as less pressure will be required and the wearing qualities will be improved. Therefore it will always be advisable to have the clutch plenty large enough for its work rather than the reverse.

With but few exceptions automobiles are equipped with ordinary conical clutches, kept in engagement by a spring or springs, and arranged to be put out of gear

automatically when the brakes are applied. In a few instances clutches of the expanding-ring form are in use, and more recently coil clutches have been adopted. The expanding-ring type would seem to offer the most advantages, as there is a minimum of end thrust to be provided for, and they are capable of being easily adjusted for wear. Probably the cheapest and most simple device for enabling the engine to be disconnected from the car is a belt in combination with fast and loose pulleys. Apart from considerations of simplicity and economy, this arrangement has the advantage of giving easy starting, is cheap and easy to repair, and, by providing a flexible transmission between the motor and the gearing, all risk of the bearings being put out of alignment is avoided. The engine and gear shafts can also be placed parallel to each other and to the driving-wheel axle, by which means the utmost efficiency will be obtained.

Belt driving alone, that is without any gearing except the driving chain or chains to the road wheels, has quite gone out of fashion. In view of the fact that there is a great demand for a reliable cheap car, it is to be questioned whether belt driving will not be revived in the near future. Probably if as much thought and attention had been devoted to the design of belt transmission gear as has been given to perfecting gear driving, belts would be more in evidence at the present time.

Truly variable speed gears have engaged the attention of many, but so far nothing really practical has resulted. The majority of designs included friction driving as part of the arrangement, and this alone is sufficient to render them impracticable. The only gear of the gradually variable type which has shown any promise of success is Hall's patent hydraulic gear, but the expense of manufacture militated against its adoption, for some time. The design has recently assumed a commercial aspect.

FRICTION CLUTCHES.

In making calculations for friction clutches of any type, it will be necessary to resolve the actual horse-power into torsional resistance at the rim of the clutch. If P be the brake horse-power to be transmitted, R the revolutions per minute, and M the twisting moment in foot-pounds, then we shall have—

$$(35) \qquad M = \frac{P33000}{R2\pi}$$

which, after reducing, may be expressed with sufficient accuracy for all ordinary purposes by—

$$(36) \qquad M = \frac{5250P}{R}$$

Now, if we make F = the mean radius of the clutch in feet, and let S = the torsional resistance, we have—

$$(37) \qquad S = \frac{M}{F}$$

To facilitate calculations, it will be preferable to express the mean radius of the clutch in inches, doing which, and substituting the value of M from equation 36, gives us—

$$(38) \qquad S = \frac{63000P}{FR}$$

No matter what design of clutch is being considered, the expression 38 remains unaltered.

The angle of the cone will depend solely upon the coefficient of friction of the materials selected and the condition of the friction surfaces. Usually the cones are of cast iron and leather, and with these materials, when both surfaces are dry, the coefficient of friction may be as high as 0·3; but to allow for the grease which generally finds its way on to the cones of a motor-car clutch, it will be safer, when making calculations, not to take the value of the

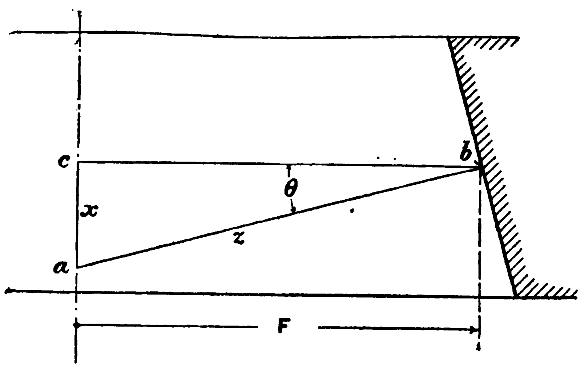


Fig. 26.

coefficient as higher than 0·2 to 0·25. Now, the coefficient of friction is the tangent of the angle of repose for the material of the clutch, and therefore for cast iron and leather the angle will be between 14° and 17°. In actual practice it is generally made 15°, and this angle will be convenient for the designer to work to, and for the machinist in manufacturing. If both surfaces of the clutch cones are of cast iron, the angle should be made 10°.

The diagram, Fig. 26, will serve to render the principle of the cone clutch perfectly clear. In the diagram ac is the axis of the clutch shaft, and $abc = \theta$ is the angle of

the cone. From any point, as c, erect a perpendicular to cb. Then, if ac represents the axial pressure forcing the cones together, ab is the resulting pressure acting in a direction perpendicular to the surface of the cone. Calling the axial pressure x, and the resulting pressure z, we have $\frac{ab}{ac} = \frac{z}{x} = \frac{1}{\sin\theta}$. If f is the coefficient of friction, it is evident that, to transmit the required power, zf must at least be equal to S. As z is equal to $\frac{x}{\sin\theta}$, we have—

$$S = zf = \frac{xf}{\sin\theta}$$

and substituting the value of S from equation 38, we arrive at $\frac{xf}{\sin\theta} = \frac{63000P}{FR}$. Hence—

$$(39) \qquad x = \frac{63000P\sin\theta}{fFR}$$

and—

$$(40) \qquad P = \frac{xfFR}{63000\sin\theta}$$

As an example, suppose we wish to ascertain what horse-power a conical clutch, with surfaces of cast iron and leather, will transmit when running at 800 revolutions per minute. We will assume the angle of the cone to be 15°, and the coefficient of friction to be 0·25.

As θ is equal to 15°, sin θ is 0·2588. The mean radius of the cone (F) is 7 inches. We will suppose the axial pressure (x) forcing the cones together to be 150 lbs., and substituting known values in equation 40, we get—

$$P = \frac{150 \times 0{\cdot}25 \times 7 \times 800}{63000 \times 0{\cdot}2588} = 13 \text{ nearly}$$

The width of the surface of the cones is arrived at from

consideration of the amount of wear likely to occur, and the allowable pressure per square inch of surface should not be greater than 50 lbs., whence we have for the width of the cone surface W—

$$W = \frac{z}{F 2\pi 50} \tag{41}$$

In the case of clutches of the expanding-ring type, all

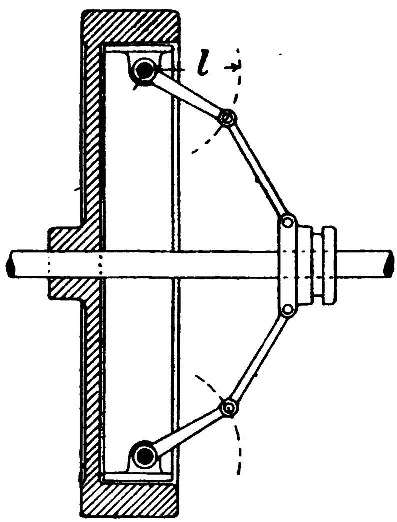


FIG. 27.

the calculations are essentially as above, but the arrangement of the levers requires consideration. Usually the ring is expanded by screws. In the majority of cases the screws are right- and left-handed, as shown diagrammatically in Fig. 27. The clutch seen in Fig. 28 (Benz-Parsifal clutch) is an example of the employment of single screws. The mechanical advantage or gain in power from the employment of the levers and screws can be found from—

(42) $$A = \frac{l2\pi}{s}$$

in which A = the gain in power, s = the pitch of the screw in inches, and l = the length of the lever in inches. This formula is applicable to clutches in which the screws are single as to the pitch, as Fig. 28. For those in which

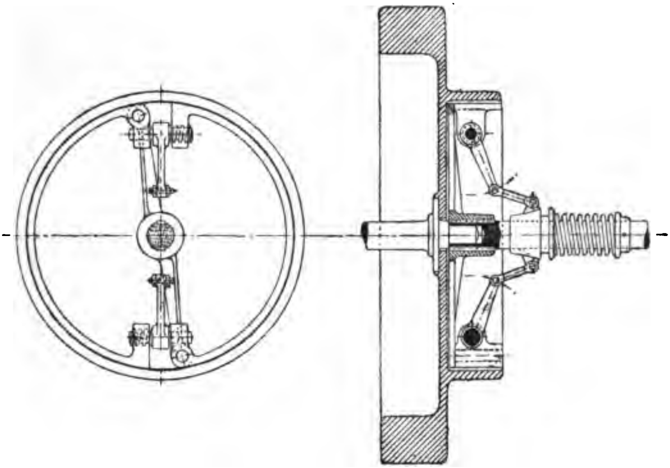


FIG. 28.

the screws are of equal but opposite pitch (Fig. 27) the expression should be halved, and so becomes—

(43) $$A = \frac{l\pi}{s}$$

From what has been said previously, it will be seen that $z = x\frac{l\pi}{s}$, and combining this with formula 39, we shall have—

(44) $$x = \frac{63000P}{FRf} \times \frac{s}{l\pi}$$

or—

(45) $$P = \frac{xfFR}{63000} \times \frac{l\pi}{s}$$

The last two formulæ are for clutches having right- and left-handed screws; for those in which only one screw or

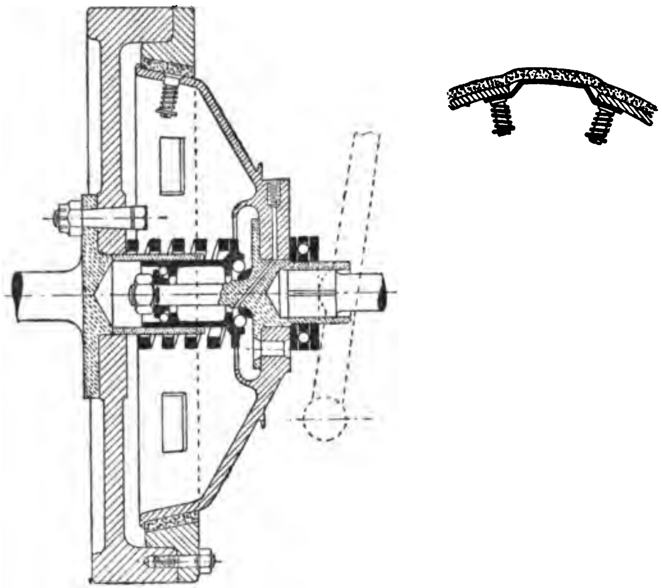


Fig. 29.

two screws of the same pitch (Fig. 28) are in question these formulæ become—

(46) $$x = \frac{63000P}{FRf} \times \frac{s}{l2\pi}$$

and—

(47) $$P = \frac{xfFR}{63000} \times \frac{l2\pi}{s}$$

It will be found that clutches designed from the foregoing considerations will be somewhat larger for a given power than is usually the case in an automobile, but the writer is convinced that the results obtained by more liberal clutch dimensions fully justify the increase in size. Automobile clutches are subjected to a lot of wear from the frequency with which they are put in and

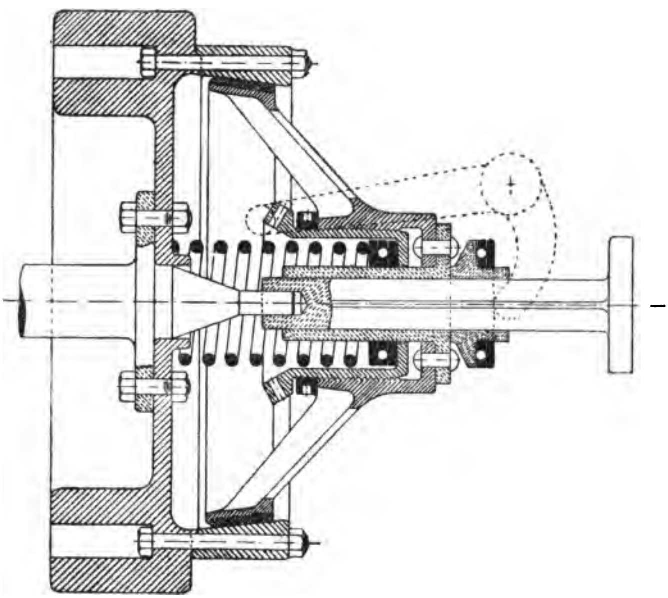


FIG. 30.

out of engagement, and this fact has received due consideration in the formulæ.

In order that the clutch shall take up its load gently and start the car without shock, it is the practice to place springs under the leather of the one cone, in places, to make the engagement gradual, and thus, by allowing a

certain amount of initial slip, render the starting easier. One arrangement of these springs is seen in the clutch shown in Fig. 29. Openings are made through the metallic portion of the cone through which bent sheet-steel pieces project. These tend to force the leather cover of the cone

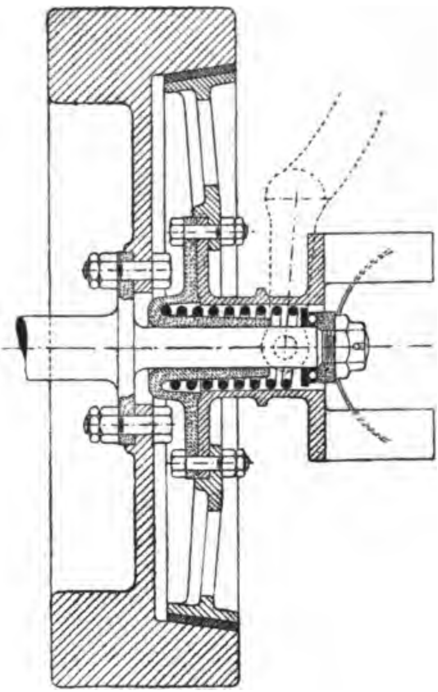


FIG. 31.

outwards by the pressure exerted by the small helical springs shown on either side of the opening.

The majority of clutches are designed so that the resultant of the axial pressure due to the spring is contained within the clutch itself, and has no effect upon the

bearings of either the crank shaft or clutch shaft when the clutch is in gear. This will be seen by reference to Fig. 29, Renault clutch; Fig. 30, Galdiator clutch; and Fig. 31, George-Richard clutch. In the Benz-Parsifal clutch (Fig. 28) the pressure of the spring reacts on the clutch shaft, but as the spring need not be anything like so strong as in the above three designs, this is not of much consequence. In the Crossley clutch (Fig. 32) the spring exercises no direct end thrust on the shafts, but there is a slight tendency for the actuating cone to be pressed back, which is resisted by the operating lever. With clutches of the designs shown in Figs. 29, 30, and 31, it is good practice to interpose a ball bearing to take the end thrust when the clutch is out of gear. Formulæ for and the method of setting out such a bearing will be found on pp. 120–123.

Clutches of the expanding-ring type, with the friction surfaces all of metal, have been used to a small extent. In this design it is usual to make the two members of the clutch of cast iron and gun-metal respectively, and to lubricate them.

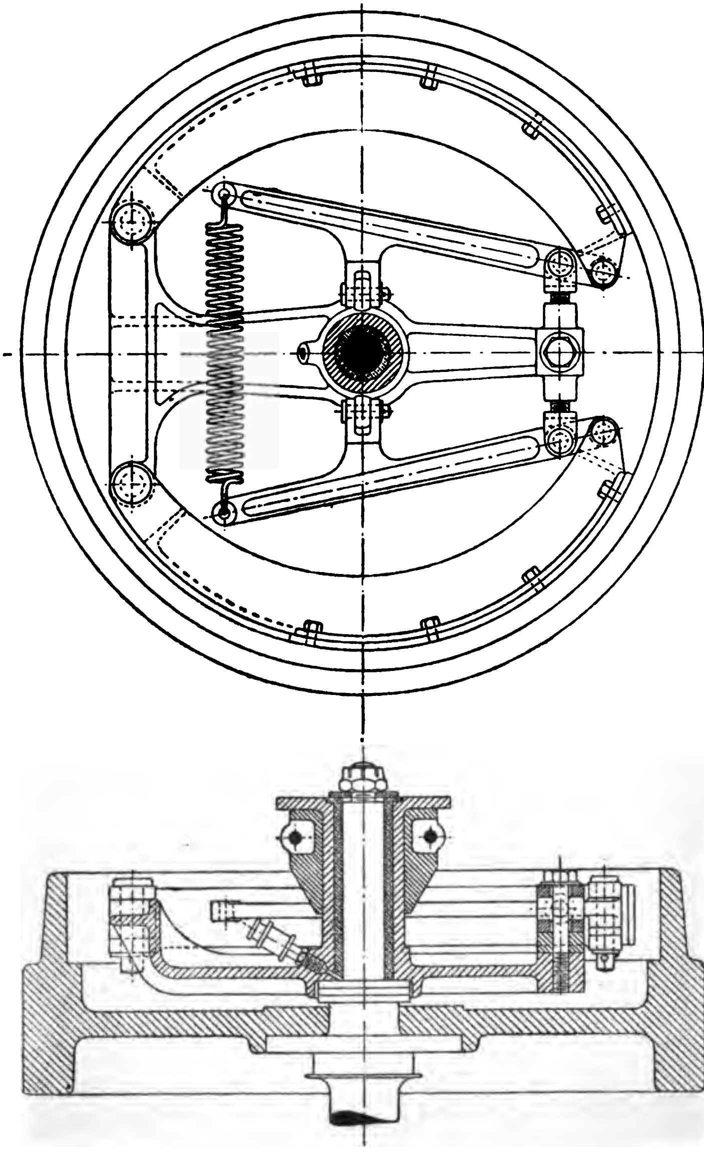


FIG. 32.

GEARING.

THE speed of an internal combustion motor may be varied between certain limits, and thereby the rate of progress of the vehicle is also affected; but as the power of the motor is directly proportional to the speed, it follows that when the maximum power of the engine is required it must run at its highest velocity. Under these conditions the car would also travel fast, but should the load on the engine, due to the weight of the car or the condition and gradient of the road, be greater than the power of the engine can deal with, it becomes necessary to adopt some form of changeable gearing whereby the rate of the vehicle can be altered without reference to the speed of the motor. In other words, the ratio of the revolutions of the engine to the revolutions of the driving wheels must be capable of alteration at will.

Many devices have been tried as change-speed gearing, but the surviving arrangement is that known as the "Panhard" design, in which a series of spur-gear wheels are made to slide into gear with another series, one at a time. Although this is most unmechanical from a theoretical point of view, its success in practice justifies its existence, and its simplicity explains its universal use. The Panhard gear has been called, and not inaptly, "clash gear." When correctly designed and constructed, and of the right material, the gear gives excellent results, but the

writer is of opinion that it would be advantageous if the gearing were made somewhat stronger than it usually is.

Gear wheels may be regarded as a development of friction gearing. In all gear wheels, whether spur, bevel, worm, or helical, there are imaginary circles, revolving in contact, known as the pitch circles, and which are analogous to friction wheels so far as speed ratios are concerned. The teeth and spaces are formed above and below these pitch circles. The distance along the pitch circle from the centre of one to the centre of the next tooth is called the circular pitch of the gear. This is also expressed as the distance occupied on the pitch circle by

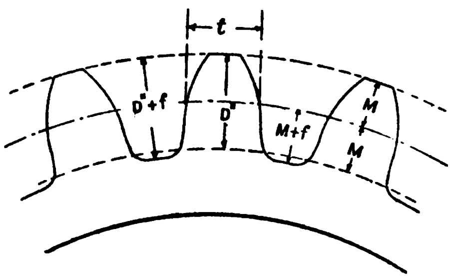


FIG. 33.

one tooth and one space. Gears with teeth of circular pitch, except in a very few cases, have inconvenient fractions in their pitch diameters, and this is apt to complicate the design, and add to the cost in manufacture. By the use of the "diametral" pitch system, the pitch diameters of the wheels can always be arranged of convenient dimensions. Circular pitch is really a measure, whereas diametral pitch is a ratio, and may be expressed as N ÷ P, where N is the number of teeth, and P the pitch diameter in inches. The nomenclature of the parts of teeth for any pitch is shown in Fig. 33, and the table, No. 9, gives the actual proportions for a number of

TABLE 9.

Table of Tooth Dimensions.
(Diametral pitch.)

Diametral pitch. P	Circular pitch in inches. P′	Thickness of tooth on pitch-line in inches. *t*	Addendum and $\frac{1''}{P}$ in inches. M	Working depth of tooth in inches. D″	Depth of space below pitch-line in inches. M + *f*	Whole depth of tooth in inches. D″ + *f*
2	1·5708	0·7854	0·5000	1·0000	0·5785	1·0785
2½	1·2566	0·6283	0·4000	0·8000	0·4628	0·8628
3	1·0472	0·5236	0·3333	0·6666	0·3587	0·7190
3½	0·8976	0·4488	0·2857	0·5714	0·3306	0·6163
4	0·7854	0·3927	0·2500	0·5000	0·2893	0·5393
5	0·6283	0·3142	0·2000	0·4000	0·2314	0·4314
6	0·5236	0·2618	0·1666	0·3333	0·1928	0·3595
7	0·4488	0·2244	0·1429	0·2857	0·1653	0·3081
8	0·3927	0·1963	0·1250	0·2500	0·1446	0·2696
9	0·3491	0·1745	0·1111	0·2222	0·1286	0·2397
10	0·3142	0·1571	0·1000	0·2000	0·1157	0·2157
12	0·2618	0·1309	0·0833	0·1666	0·0964	0·1798
14	0·2244	0·1122	0·0714	0·1429	0·0826	0·1541
16	0·1963	0·0982	0·0625	0·1250	0·0723	0·1348
18	0·1745	0·0873	0·0555	0·1111	0·0643	0·1198
20	0·1571	0·0785	0·0500	0·1000	0·0579	0·1079

different pitches. The comparative sizes of teeth of the most usual diametral pitches may be seen at a glance from Fig. 34. The following formulæ will be found useful in connection with calculations of gear-wheel dimensions and velocity ratios.

Formulæ for gears of diametral pitch—

Let P = diametral pitch.
D′ = pitch diameter.
D = whole diameter of the wheel blank.
N = number of teeth in the gear.
V = velocity in revolutions per minute.
d' = pitch diameter.
d = whole diameter of the pinion blank.
n = number of teeth in the pinion.

Let v = velocity in revolutions per minute.
a = centre distance.
b = number of teeth in both wheels.
t = width of tooth, or space, or pitch-line.
D'' = working depth of tooth.
f = amount added to working depth for clearance.
$D'' + f$ = whole depth of tooth
P' = circular pitch.
π = a constant = 0·3146

Then, for a single wheel—

$$P = \frac{N+2}{D} \qquad \text{or,} \quad P = \frac{N}{D'}$$

$$D' = \frac{D \times N}{N+2} \qquad \text{or,} \quad D' = \frac{N}{P}$$

$$N = PD' \qquad \text{or,} \quad N = PD - 2$$

$$D = \frac{N+2}{P} \qquad \text{or,} \quad D = D' + \frac{2}{P}$$

$$t = \frac{1{\cdot}57}{P} \qquad D'' = \frac{2}{P} \qquad f = \frac{t}{10}$$

$$P' = \frac{\pi}{P} \qquad \text{and} \qquad P = \frac{\pi}{P'}$$

For a pair of wheels—

$$b = 2aP \qquad n = \frac{bv}{v+V} \qquad N = \frac{nv}{V}$$

$$n = \frac{NV}{v} \qquad N = \frac{bv}{v+V} \qquad n = \frac{PD'V}{v}$$

$$V = \frac{nv}{N} \qquad v = \frac{NV}{n} \qquad v = \frac{PD'V}{n}$$

$$D = \frac{2a(N+2)}{b} \qquad d = \frac{2a(n+2)}{b} \qquad a = \frac{b}{2P}$$

$$D' = \frac{2av}{v+V} \qquad d' = \frac{2aV}{v+V} \qquad a = \frac{D'+d'}{2}$$

16 P

P

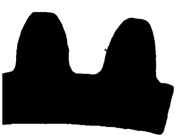
P

face p. 92.

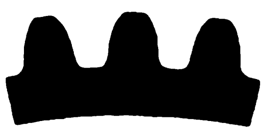
Mod. 7

Mod. $6\frac{1}{2}$

Mod. 4



Mod. $3\frac{3}{4}$



Mod. $1\frac{1}{2}$

Mod. $1\frac{1}{4}$

Mod. 1

In France "module" pitch is almost universally used, and is gradually coming into use in this country. This is a modification of diametral pitch, and has the advantage of only requiring measurements in millimetres and such fractions thereof as 0·25, 0·5, and 0·75. Diametral pitches call for unusual divisions of the inch, such as sevenths, ninths, elevenths, etc., which are apt to be confusing to both the designer and the workman. The module corresponds to the height of the tooth above the pitch-line, and is the pitch diameter in millimetres divided by the number of teeth. Conversely the pitch diameter is equal to the module multiplied by the number of teeth. The following formulæ will enable the designer to calculate all necessary dimensions of gears of module pitch, and the table, No. 10, will facilitate the work. Fig. 35 shows at a glance the comparative sizes of module-pitch teeth from Mod. 1 to Mod. 12.

Formulæ for gears of module pitch—

Let M = module in millimetres.
D' = pitch diameter in millimetres.
D = whole diameter in millimetres.
N = number of teeth.
D'' = working depth of teeth.
t = thickness of teeth on pitch-line.
f = amount added to working depth for clearance.
C = circular pitch in millimetres.

Then—

$$M = \frac{D'}{N} \qquad \text{or, } M = \frac{D}{N+2}$$

$$D' = NM \qquad D = (N+2)M$$

$$N = \frac{D'}{N} \qquad \text{or, } N = \frac{D}{M} - 2$$

$$D'' = 2M \qquad t = M1{\cdot}5708 \qquad f = \frac{M1{\cdot}5708}{10}$$

$$C = M \times 3{\cdot}1416 \qquad M = C \div 3{\cdot}1416$$

TABLE 10.

Module Pitch.

(Tooth dimensions in millimetres.)

Module.	Circular pitch, millimetres.	Addendum, millimetres.	Total height of tooth, millimetres.	Corresponding English diametral pitch. No.
1	3·14	1·0	2·16	25·400
1¼	3·93	1·25	2·7	20·320
1½	4·71	1·5	3·23	16·933
1¾	5·5	1·75	3·77	14·514
2	6·28	2·0	4·31	12·700
2¼	7·07	2·25	4·85	11·288
2½	7·86	2·5	5·4	10·160
2¾	8·63	2·75	5·93	9·236
3	9·42	3·0	6·47	8·466
3¼	10·2	3·25	7·0	7·81
3½	11·0	3·5	7·55	7·257
3¾	11·77	3·75	8·090	6·773
4	12·57	4·0	8·63	6·350
4¼	13·35	4·25	9·17	5·708
4½	14·14	4·5	9·71	5·644
4¾	14·92	4·75	10·24	5·347
5	15·71	5·0	10·78	5·080
5¼	16·49	5·25	11·33	4·838
5½	17·28	5·5	11·86	4·618
6	18·86	6·0	12·94	4·233
6½	20·41	6·5	14·02	3·907
7	22·0	7·0	15·1	3·628
8	25·14	8·0	17·26	3·175
9	28·27	9·0	19·41	2·822
10	31·41	10·0	21·57	2·540
11	34·56	11·0	23·72	2·309
12	37·7	12·0	15·88	2·117

To design a change-speed gear for a car the data required are (*a*) normal speed of the motor, (*b*) number of speeds required, (*c*) value of these speeds in miles per hour, and (*d*) diameter of the driving wheels. By the aid of table No. 11, p. 95, the revolutions per minute of the driving wheels for any given number of miles per hour can be readily obtained by multiplying the number in the

fourth column, opposite the given diameter of wheel, by the number of miles per hour desired. Thus a wheel of 32 inches diameter, running at a speed of 12 miles per hour, will revolve at $10{\cdot}5 \times 12 = 126$ revolutions per

TABLE 11.

Driving-wheel Diameters and Speeds.

Diameter in inches.	Circumference in inches.	Revolutions per mile.	Revolutions per minute = 1 mile per hour.
24	75·39	863·2	14·38
25	78·54	806·7	13·45
26	81·68	775·7	12·92
27	84·82	747·0	12·45
28	87·96	720·3	12·00
29	91·10	695·5	11·58
30	94·24	672·3	11·20
31	97·39	650·5	10·92
32	100·53	630·2	10·5
33	103·67	611·1	10·18
34	106·81	588·5	9·8
35	109·95	576·2	9·6
36	113·09	560·2	9·33
37	116·23	545·5	9·09
38	119·38	530·7	8·84
39	122·52	517·1	8·61
40	125·66	504·2	8·4
41	128·8	492·0	8·2
42	131·94	480·0	8·0
43	135·08	475·5	7·92
44	138·23	451·1	7·51
45	141·37	448·1	7·46
46	144·51	438·4	7·3
47	147·65	422·3	7·03
48	150·79	420·0	

minute. It will be necessary to assume the ratio of the bevel or sprocket wheels by which the power is transmitted from the change-speed gear to the road wheels, and this should be such that when the car is running on the highest gear the drive will be direct from engine to road wheels with only the speed reduction due to the bevel or chain

gear. Taking the above example, and assuming the wheels to be driven by chain gear with sprockets having a ratio of 4 to 1, we obtain a speed of $126 \times 4 = 504$ (say 500) revolutions per minute for the driving sprockets. Usually the centre distance of the driving and driven shafts in the gear box is limited, and the problem is to find the pitch diameters for a pair of gear wheels to run at a given speed ratio, with the centre distance fixed. The rule is, "divide the centre distance by the sum of the terms of the ratio, find the product of twice the quotient by each of the terms separately, and the two products thus obtained will be the pitch diameters of the two gears."

Again taking the above example, and assuming the normal speed of the motor to be 1000 revolutions per minute, and the centre distance of the shafts to be fixed at 6 inches, we require to know the pitch diameters of a pair of gears to revolve at 1000 and 500 revolutions per minute respectively. Adding the terms of the ratio, $10 + 5 = 15$, and dividing the centre distance, 6 inches, by the sum, we obtain 0·4, and $0{\cdot}4 \times 2 = 0{\cdot}8$, which, multiplied by each of the terms of the ratio, gives 8 inches and 4 inches as the required diameters.

The pitch of the teeth and the width of the gear are determined by (*a*) the power to be transmitted, (*b*) the velocity of the pitch-line in feet per minute, and (*c*) the material used for the gear. Rules and formulæ for the strength of gear wheels are numerous, and the results obtained by them vary very considerably; so much so that when a gear wheel fails from any cause, it is not difficult to find a rule to justify one for having used the wheel. The formulæ used and recommended by the writer take into account the safe fibre stress, with the tooth considered as a beam loaded at one end and supported at the other. The working load for teeth of 1 inch width and a fibre stress of 1000 lbs. per square inch have been

calculated for diametral pitches from 3 to 12, and will be found in the table below. To obtain the safe working load on a gear wheel the formula is—

$$\text{(48)} \qquad L = t \times f \times s \times m$$

where L = the safe load in pounds, t = the tabular number

TABLE 12.

VALUES OF L FOR TEETH OF 1-INCH FACE, AND 1000 LBS. PER SQUARE INCH FIBRE STRESS.

No. of teeth in wheel.	Diametral pitch.												
	3	3¼	3½	3¾	4	5	6	7	8	9	10	11	12
12	70	65	60	56	52	42	35	30	26	23	21	19	17
13	73	68	63	59	55	44	37	31	27	24	22	20	18
14	75	69	65	60	56	45	38	32	28	25	22	20	19
15	79	73	67	63	59	47	39	33	28	26	23	21	19
16	81	74	69	64	60	48	40	34	30	27	24	22	20
17	84	77	72	67	63	50	42	36	31	28	25	23	21
18	87	80	75	70	65	52	43	37	33	29	26	24	21
19	91	84	78	73	68	54	45	39	34	30	27	25	22
20	94	87	81	75	71	57	47	40	35	31	28	25	23
21	96	89	83	77	72	58	48	41	36	32	28	26	24
23	98	91	84	79	74	59	49	42	37	33	29	26	24
25	102	94	87	81	76	61	51	43	38	34	30	27	25
27	104	97	90	84	78	63	52	45	39	35	31	27	26
30	106	99	91	85	80	64	53	46	40	36	32	29	26
34	109	101	94	88	82	66	54	47	41	36	33	29	27
38	112	103	96	90	84	67	56	48	42	37	33	30	28
43	115	105	99	92	86	69	57	49	43	38	34	31	28
50	117	108	100	94	88	70	58	50	44	39	35	32	29
60	119	110	102	95	89	71	59	51	44	40	35	32	29
75	121	112	104	97	91	73	61	52	45	40	36	33	30
100	124	114	106	99	93	74	62	53	46	41	37	33	31
150	126	116	108	100	94	75	63	54	47	42	37	34	31
300	128	118	110	102	96	77	64	55	48	43	38	35	32
Rack	130	120	112	104	97	78	65	56	49	44	39	36	33

corresponding to the pitch and number of teeth in the gear, f = the width of face in inches, s = the speed

coefficient from the diagram Fig. 35A, and $m =$ the safe fibre stress obtained from the table below.

TABLE 13.

FIBRE STRESSES.

Materials.	Ultimate.	Safe.
Cast iron	22,000	8,000
Gun-metal	34,000	11,000
Phosphor bronze	50,000	16,000
Cast steel	60,000	20,000
Forged steel	65,000	25,000

For example, suppose we have a mild-steel gear of 30 teeth, 6 diametral pitch, 2 inches wide on the face, and running at a pitch-line velocity of 600 feet per minute. Substituting in 48, we have—

$$L = 53 \times 2 \times 0{\cdot}4 \times 25 = 1060 \text{ lbs.}$$

The value of m, although given as 25,000 in the table, for mild steel, is only taken as 25 in making calculations, as the tabular number is already calculated for 1000 lbs., and therefore only requires to be multiplied by 25 to equal the safe fibre stress.

To ascertain the horse-power the gear will transmit the formula is—

$$(49) \qquad \text{H.P.} = \frac{L \times r}{33000}$$

where $L =$ the safe load as found by 48, and $r =$ the pitch-line velocity in feet per minute. Taking the wheel in the above example, and substituting known values in 49, we have—

$$\text{H.P.} = \frac{1060 \times 600}{33000} = 19 \text{ about}$$

The above rules will apply equally well for bevel gears, provided that the *mean* pitch, *mean*-pitch diameter, and the velocity of the *mean*-pitch line are taken. For gears of module or circular pitch the equivalent diametral pitch should first be obtained from table No. 10, p. 94.

Pitch-line velocities of more than 2000 feet per minute should be avoided as tending to cause considerable noise, however well the teeth may be cut. At this speed the

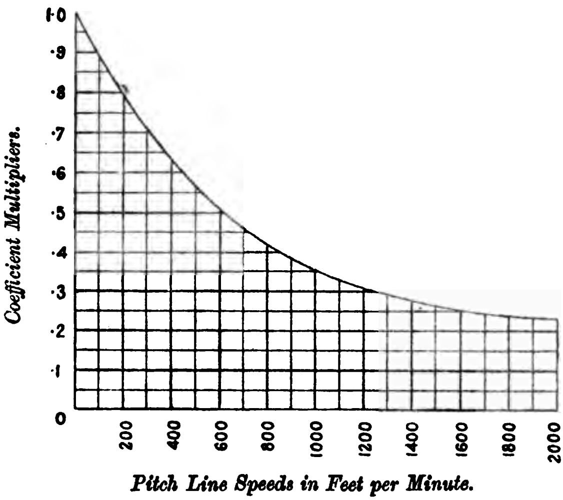


FIG. 35A.

gears require very accurate work, and also care in assembling to make them run with anything like silence. As a general rule, the higher the speed the finer the pitch should be, and fine pitches always run smoother than coarse.

Gears made from special materials such as rawhide, vulcanized fibre, and Unica fibre, should have a liberal factor of safety, and the writer can recommend either of the two following formulæ:—

$$(50) \qquad \text{H.P.} = \frac{p \times d \times f \times r}{1000}$$

where p = circular pitch in inches, d = pitch diameter in inches, f = width of face in inches, and r = revolutions per minute.

$$(51) \qquad \text{H.P.} = x \times p \times d \times r \times f,$$

where $x = 0.0131$, p = circular pitch in inches, d = pitch diameter in feet, and the other factors are as above. No reliable figures are available for the strength of rawhide and fibre as applied to gear-wheel construction, but it may be safely taken as fully equal to that of cast iron, and with a higher elastic limit. In weight, rawhide, vulcanized and Unica fibre, may all be reckoned about the same, *i.e.* 20 cubic inches per pound, or 0·05 lb. per cubic inch. Rawhide gears should always have a metal plate on each side to prevent the layers of hide from spreading, such plates being secured by riveting right through the hide and both plates. Unica and vulcanized fibre wheels do not require plates, but their strength is increased by putting a few rivets transversely through them. On no account should oil or grease be allowed to get upon rawhide wheels, as they will be softened, and probably spoilt, by it, but fibre wheels are not affected. The only lubricants for rawhide, if required, are French chalk or graphite. Printing ink is useful as a lubricant when hide pinions are required to run in a damp atmosphere. The width of face of a rawhide gear between the side plates should always be slightly greater than that of the wheel with which it meshes, so that the metal side plates do not come in contact with the wheel. Rawhide and fibre gears are specially useful where very high speeds have to be dealt with, owing to the freedom from noise and vibration

secured by their use; hence they are more frequently used for electric than petrol automobiles.

Bevel gears are a common feature of the transmission gearing of present-day cars; nevertheless, their use is not recommended. The writer is of opinion that bevel-gear driven live axles will before long be discarded, for all but small cars, in favour of sprocket wheels and chains. In some designs the use of bevel gearing is unavoidable, in which case the end thrust must be provided for. A ball or roller thrust bearing is usually employed. The pro-

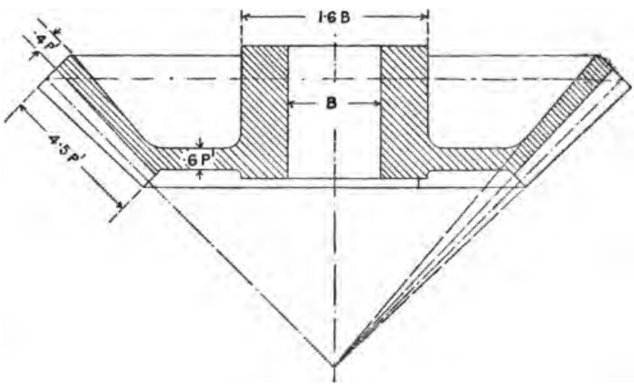


Fig. 36.

portions given for bevel wheels in Fig. 36 are based on average practice. There will be no gain in making the width of the face greater than as given; the pitch of the teeth at the smaller diameter would become too fine to be of much service.

Worm gearing finds only limited application in an automobile, its chief use being for irreversible steering gears. In one or two designs worm gears are employed in place of chain or bevel gearing to drive the back axle. When used for this purpose the angle of the worm thread

should be 45°, in order that the gear may act equally well in either direction, *i.e.* either the worm or the wheel may be the driver. It is not recommended by the writer that worm gearing should be used for driving the axle. It is

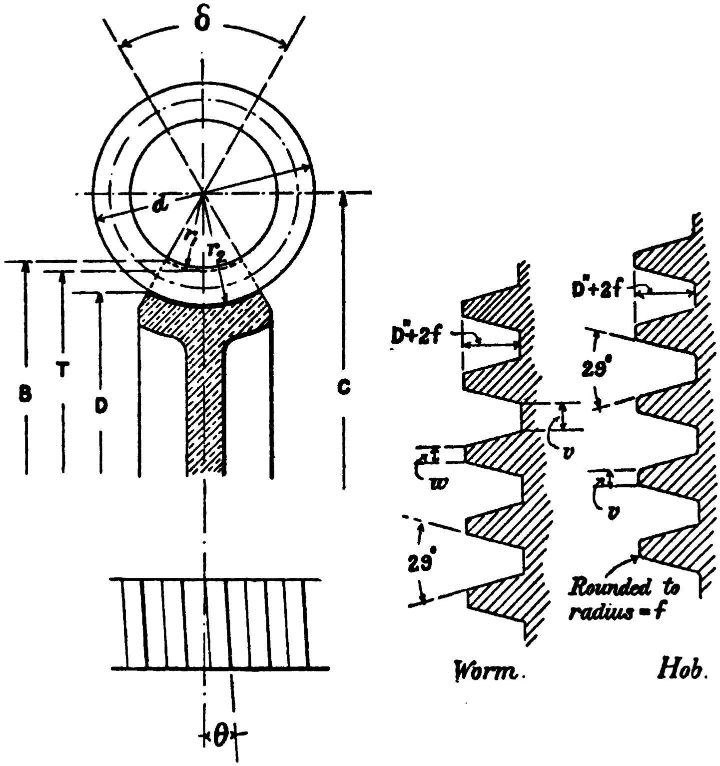


FIG. 37.

well known that the frictional losses with this gear are very high, but this will be no detriment to its use for steering purposes.

Fig. 37 will give the designer all necessary proportions

for worm gears, and the following formulæ will enable him to calculate dimensions:—

Formulæ for worm gearing—

Let L = lead of worm.
N = number of teeth in wheel.
m = threads, or turns, per inch in worm.
d = diameter of worm.
d' = diameter of worm hob.
T = throat diameter of wheel.
B = diameter of wheel blank to sharp corners.
o = width of slots in hob.
l = width of hob teeth at bottom.
b = pitch circumference of worm.
v = width of worm-thread tool at end.
w = width of worm thread at top.
P = diametral pitch.
P' = circular pitch.
s = addendum, or module.
t = thickness of tooth on pitch-line.
t^n = normal thickness of tooth on pitch-line.
f = clearance.
D'' = working depth of tooth.
$D'' + f$ = whole depth of tooth.
θ = angle of teeth of wheel with axis.

Then—

For a single-thread worm, $L = P'$; for a double-thread, $L = 2P'$; for a triple-thread, $L = 3P'$; and so on.

$$L = \frac{1}{m} \qquad P' = \frac{\pi T}{N+2} \qquad D = \frac{NP'}{\pi} \qquad \text{or, } D = \frac{N}{P}$$

$$T = \frac{N}{P} + 2s \qquad b = \pi(d - 2s) \qquad t^n = t\cos\theta \qquad r' = \frac{d}{2} - 2s$$

$$C = \frac{D+d}{2} - s \qquad B = T + 2\left(r' - r'\cos\frac{\delta}{2}\right)$$

$$o = \frac{0{\cdot}335\text{P}'}{2} + \tfrac{1}{8}'' \qquad l = \text{D}'' + 2f + \tfrac{1}{8}''$$

$$d' = d + 2f \qquad v = 0{\cdot}31\text{P}' \qquad w = 0{\cdot}335\text{P}'$$

The angle δ is usually made 60° to 90°.

The thread shown in Fig. 37 is the Acme 29° thread, and the dimensions of the thread for several sizes are given in the table below—

TABLE 14.

TABLE OF THREAD PARTS.

No. of threads per inch. Linear.	Depth of thread.	Width at top of thread.	Width at bottom of thread.	Space at top of thread.	Thickness at root of thread.
1	0·5100	0·3707	0·3655	0·6293	0·6345
$1\frac{1}{3}$	0·3850	0·2780	0·2728	0·4720	0·4772
2	0·2600	0·1853	0·1801	0·3147	0·3199
3	0·1767	0·1235	0·1183	0·2098	0·2150
4	0·1350	0·0927	0·0875	0·1573	0·1625
5	0·1100	0·0741	0·0689	0·1259	0·1311
6	0·0933	0·0618	0·0566	0·1049	0·1101
7	0·0814	0·0529	0·0478	0·0899	0·0951
8	0·0725	0·0463	0·0411	0·0787	0·0839
9	0·0655	0·0413	0·0361	0·0699	0·0751
10	0·0600	0·0371	0·0319	0·0629	0·0681

Driving chains are generally of the roller variety, block chains not being much employed, though a few years ago they were much in evidence. Roller chains absorb less energy than block, and it is possible, with a roller-chain drive, under favourable conditions, for the efficiency of the drive to be as high as 98 per cent.

The sprocket wheels for chains of the block type may be designed from the following formula. Let N = the number of teeth, p = the pitch, d = the diameter of the round part of the chain block (usually $0{\cdot}325p$), D = the distance from centre to centre of the rivet holes in the

block (usually $0{\cdot}4p$), and C = the distance from centre to centre of the rivet holes in the side plate (generally $0{\cdot}6p$); then—

$$(52) \qquad \text{Pitch diameter} = \frac{C}{\sin \beta}$$

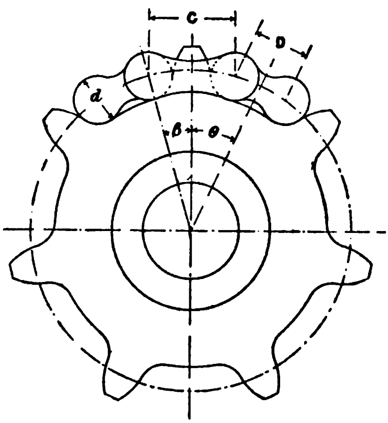


FIG. 38.

Fig. 38 illustrates a sprocket wheel for block chain, and the factors above appear thereon. The angle θ is equal to $\frac{180°}{N}$, the value of which for any number of teeth up to 100 may be readily ascertained from table No. 46, p. 170. Also $\tan \beta \quad \dfrac{\sin \theta}{\frac{D}{C} + \cos \theta}$.

The diameter over the points of the teeth may be taken as the pitch diameter plus d, and the diameter at the root of the teeth, which is of great importance, may be the pitch diameter minus d.

For roller-chain sprockets the diameter of the pitch circle may be calculated from—

$$(53) \qquad D = \frac{p}{\sin\left(\frac{180°}{N}\right)}$$

where D = the pitch diameter, or the diameter of the circle passing through the centres of the rollers; p = the pitch in inches; N = the number of teeth; and d = the diameter of the roller (usually $0{\cdot}5p$). The diameter over the points of the teeth will be $D + d$, and at the root of the teeth $D - d$. The table No. 46, p. 170, will be useful in calculating the dimensions of roller-chain wheels. Values of $\frac{360°}{N}$, $\frac{180°}{N}$, and $\sin \frac{180°}{N}$ will be found tabulated for numbers from 1 to 100.

The design of the actual tooth outlines is a matter which does not concern an automobile designer; the cutters employed on the gear-cutting machines take care of this. It may be mentioned that the involute tooth curve is better suited to motor-car speed gears than the cycloidal curve, as it allows the shaft centres to be slightly varied without materially affecting the working of the gears. Teeth cut to cycloidal curves render it imperative that the shaft centres of the gears shall be exactly the correct distance apart, the slightest variation causing the gears to be noisy and to wear badly. If the gear-wheel shafts can be placed the correct distance apart to commence with, and be always maintained so, it will be found that the cycloidal tooth will give quieter running, but in practice it is generally found that the shaft centre distance will be altered by reason of the wear of the bearings, and in the case of gear boxes of aluminium the alloy has been known to stretch under the pressure exerted by the gears, and thus increase the distance between the shaft centres.

With a change-speed gear of the Panhard type it is usual to provide a square shaft for the sliding train of

gears. The writer advocates a round shaft with a couple of sunk feathers in place of this, as, apart from the cost of manufacture of a square shaft, and of accurately fitting the gear sleeve to the shaft, it will be found very difficult to ensure that the gear wheels shall run absolutely true. Even when quite true to commence with, very little wear will throw the gears out of truth and give rise to noise. With a round shaft and a round hole through the gear sleeve, the work can be ground after hardening and a perfect fit guaranteed.

Gear wheels with less than twelve teeth should never be used, and it will be preferable to make fifteen teeth the minimum.

A form of speed-reducing gear which is coming into favour is that known as the "crypto" or epicyclic gearing. Though not by any means new, the application of this gear to automobiles has only recently received much attention, and this is chiefly because of its success in small cars of American manufacture. When properly designed for the work it has to do, this gear is very satisfactory, owing to the smoothness of running and the ease with which the load is taken up. Some years ago the writer applied this gear to a motor bicycle, reducing the speed in the ratio of 4 to 1, and driving from the gear to the road wheel by means of a chain. The principal advantages derived from the use of the gear in this connection were that it allowed a certain amount of slip, sufficient to avoid breakage of the driving chain, and it permitted the engine to be disconnected from the road wheel at will, thus facilitating riding in traffic. Also the motor could be started before mounting the machine with a starting handle in the usual manner, and the gear only allowed to act when the bicycle was under way, the gear coming into action very gradually. The usual form of this gear is seen in Fig. 39. The pinion A is the driver, being secured

to the engine shaft usually, or connected thereto by a chain drive. The four planetary pinions C revolve freely on their spindles, which are fixed into the plate D. To the plate D is secured the sprocket wheel which drives the back wheels of the car. When the pinion A is revolved, and the internally toothed ring B is held from revolving, the planetary pinions C are caused to run around the ring B and carry the plate D with them. The ratio of speed reduction with this gear is directly as the ratio of the number of teeth in the driving pinion A to the number of teeth in the ring B *plus one.* Thus in Fig. 39 the pinion A has 18 teeth, and the ring B has 54 teeth, hence the plate D will revolve at one-fourth the speed of A, *i.e.* the ratio of 18 to 54 + 1. It is to be noted that the spindles of the planetary pinions can only be equally spaced when the number of the pinions can be divided into the number of teeth in both the pinion A and the ring B without a remainder. Thus in the gear seen in Fig. 39 it would be impossible to have four planetary pinions equally spaced around A, because the number of teeth in A and B cannot be divided by 4 without a remainder.

The pitch of the teeth in an epicyclic gear can be made much finer than would be safe with an ordinary reduction gear, as there are more teeth in action to take the load. It would be safe to reckon that in the gear, Fig. 39, there will never be less than three teeth in action at once, where in an ordinary pair of wheels the probability is that only one tooth would at times take the whole load. It is probably due to this that the epicyclic gear runs with such smoothness.

When applied to an automobile, it is usual to make the reversing gear of similar design to the slow-speed forward gear, but with this difference, that the internally toothed ring B becomes the driven member of the gear,

the plate D being held stationary. In this case the ring

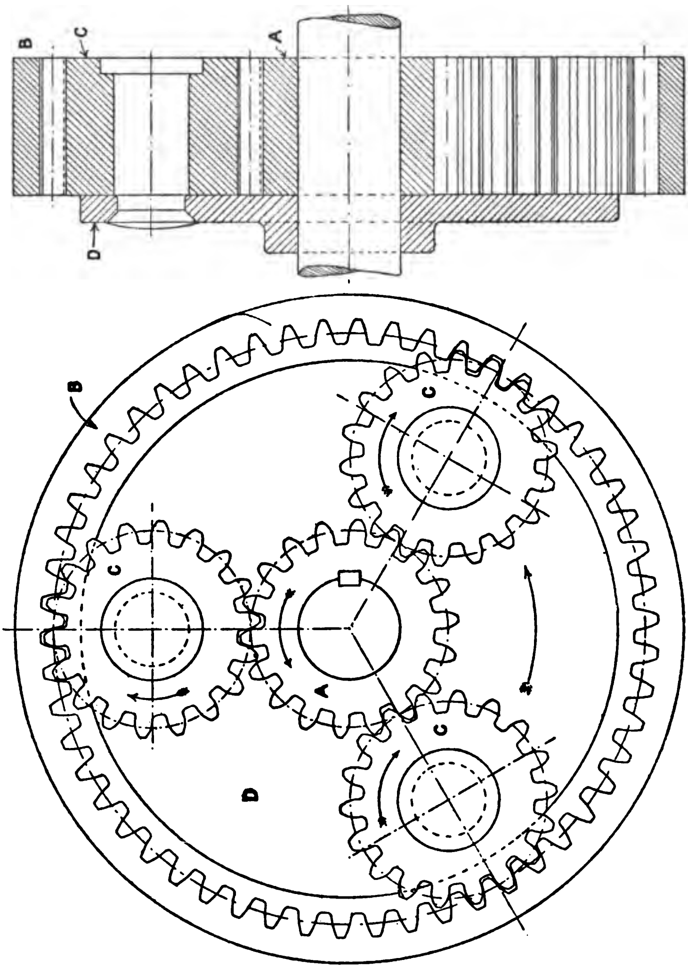


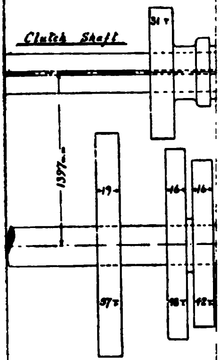
FIG. 39.

B will revolve in the opposite direction to the pinion A, and the ratio of reduction will be directly as the numbers

of the teeth in A and B, the planetary pinions merely acting as intermediate gears without any influence on the speed ratio. When the highest speed is required it is usual to employ a clutch and cause the whole of the gear to revolve at the same speed as the engine shaft. That is to say, the gear is idle, no teeth being exchanged, and the only reduction in speed between the motor and the road wheels is that due to the ratio of the number of teeth in the sprocket wheels used.

Epicyclic gearing has been employed in place of a main clutch in some cars, the driving pinion A being secured to the motor shaft and the plate D to the speed-gear shaft. By checking the ring B slowly the load is taken by the engine very gradually, and the car can be started without any shock. A further advantage is that there is no end thrust on either the motor or the gear shafts as with a conical clutch. The gear should be so designed as to be capable of being filled with lubricant, when the wear will be but slight. By employing gear wheels with the teeth cut at a slight angle, *i.e.* helical gears, epicyclic gearing can be made practically noiseless in action.

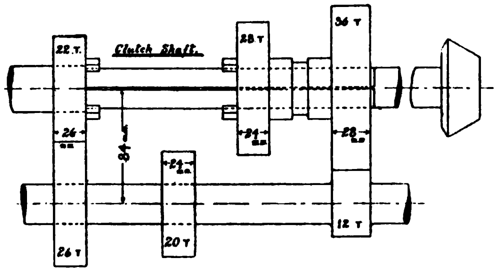
The annexed diagrams of speed gears show the principal dimensions of the gears used in some well-known and successful cars. The face width of the gears is to some extent reduced by the teeth being bevelled off at one or both ends, to facilitate the meshing of the gears. In most cases this loss of effective width amounts to a full $\frac{1}{8}$ inch for each end bevelled, and should be allowed for when calculating the strength of the gear.



CHARRON, GIR

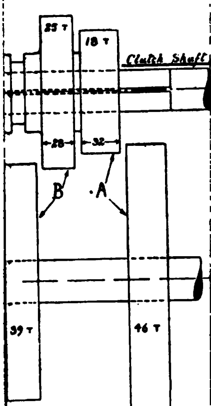
ylinders, 110 mm. bor

7 pitch ;



DECAUVILLE.

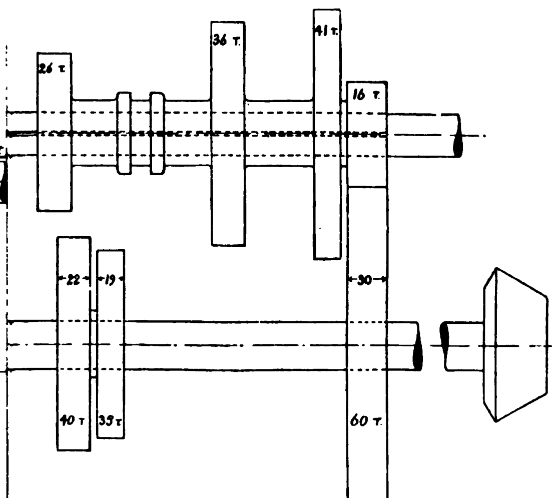
4 cylinders, 100 mm. bore × 110 mm. stroke. All gears, module $3\frac{1}{2}$ pitch.



ERCÉDÈS.

× 125 mm. stroke. G

C and D, module 4 p



PANHARD.

p. bore × 140 mm. stroke. All gears, module 4 pitch.

[To face p. 110.

BRAKES.

PROBABLY no part of an automobile requires more careful design than the brakes, in view of the probability that efficient stopping power may on occasion prevent loss of life. A high factor of safety is desirable, and in calculating strengths the weakest places should receive chief consideration.

The duty of the brakes is to dissipate the energy stored in the moving vehicle, and in the shortest possible time. Hence, before we can calculate the strength required for any part of the brake gear, the energy contained in the moving mass must be ascertained.

Denoting the energy in foot-pounds by E, the weight of the vehicle in pounds by W, and the velocity in feet per second by s, we have—

$$\text{(54)} \qquad E = \frac{Ws}{2g}$$

g representing the acceleration due to gravity = 32·2 feet per second. It will be more convenient to modify the expression to agree with miles per hour, as the speed of motor vehicles is generally expressed in these terms. Therefore, as 1 mile is = 5280 feet, and 1 hour is = 3600 seconds, 1 mile per hour is $= \frac{5280}{3600} = 1{\cdot}466$ feet per

second. Hence for 1 mile per hour we have, substituting in 54—

$$E = \frac{W \times 1{\cdot}466^2}{64{\cdot}4} = W \times 0{\cdot}0334$$

and putting S = miles per hour, the expression 54 becomes—

$$(55) \qquad E = WS^2 \times 0{\cdot}0334$$

As an example, we will assume a vehicle weighing, including its load, 16 cwt., running at a speed of 15 miles per hour. Ignoring road resistance, and substituting known values in 55, we obtain—

$$E = 1792 \times 225 \times 0{\cdot}0334 = 13466 \text{ foot-pounds}$$

A factor of importance to be considered when calculating the stopping power of the brakes is the coefficient of friction between the tyre of the wheel and the road surface. For want of precise information on this head we may assume this coefficient to be 0·4 for iron tyres and 0·7 for rubber tyres. These values will vary according to the condition and material of the road surface. For wood and asphalte roads, when the surface is greasy, the coefficients should be taken as not more than one-half the above values. On the majority of automobiles the brakes only act upon two of the wheels, usually the driving wheels. Therefore we shall require to know the proportion of the total weight of the vehicle carried by the wheels to which the brakes are applied. For the example selected above we will assume this to be one-half of the total weight = 896 lbs. Then for the minimum distance in which the car can be stopped we have the expression—

$$(56) \qquad L = \frac{WS^2 \times 0{\cdot}0334}{k \times w} = \frac{E}{k \times w}$$

in which L = the minimum distance in which the car can be stopped, k = the coefficient of friction between the tyre and the road, and w = the weight in pounds carried by the wheels to which the brakes are applied; the other factors being as before. Substituting known values in 56, we have—

$$L = \frac{13466}{0{\cdot}4 \times 896} = 37{\cdot}5 \text{ feet for iron tyres}$$

$$L = \frac{13466}{0{\cdot}7 \times 896} = 21{\cdot}4 \text{ feet for rubber tyres}$$

By dividing the energy E stored in the moving car by the distance L, we obtain the mean resistance required on the periphery of the tyres. Thus—

$$(57) \qquad P = \frac{WS^2 \times 0{\cdot}0334}{L} = \frac{E}{L} = kw$$

In our example $P = \frac{13466}{37{\cdot}5}$ = say 360 lbs. for iron tyres, or $P = \frac{13466}{21{\cdot}4}$ = say 630 lbs. for rubber tyres; one-half of these amounts being required on each of the two wheels. From this it will be seen that the coefficient of friction for rubber tyres, being greater than for iron, allows of a stronger breaking effort being applied without skidding the wheels, and hence the car is stopped in a much shorter distance.

When band brakes are used, their diameter is of necessity less than that of the road wheels, and the pull on the brake rods will be increased in inverse proportion to the diameter of the road wheel and brake drum. Thus—

$$(58) \qquad p = kw\frac{D}{d}$$

in which D = the diameter of the road wheel, d = the diameter of the brake drum, p = the pull on the brake rod; and the other factors as before. Assuming the road wheels of the car in the example to be 30 inches diameter, and the brake drums 10 inches diameter, and rubber tyres on the wheels, we have, from 58, $p = 630 \times \frac{30}{10} = 1890$ lbs. for the pair of brake bands, or 945 lbs. mean pull on each. In the case of sudden applications of the brakes, such as cause the wheels to skid, this pull of 945 lbs. will be much exceeded.

For shoe brakes, acting directly on the tyres, the pressure can be found by dividing the mean resistance P by the coefficient of friction, or $\frac{P}{k}$, which in our example gives $\frac{315}{0 \cdot 7} = 450$ lbs. on each tyre.

In all the above calculations we have assumed the car to be travelling on a level road. Gradients will tend to lengthen or shorten the distance in which the car can be stopped, according to whether it is an up or down grade. Down grades will reduce the resistance P, and may be expressed as a fraction $= f$. The distance L in which the car can be stopped going downhill may then be calculated from—

$$(59) \qquad L = \frac{WS^2 \times 0 \cdot 0334}{(kw) - (W \div f)}$$

Substituting known values from our example, and assuming a gradient of 1 in 20, we have—

$$L = \frac{13466}{(0 \cdot 7 \times 896) - (1792 \div 20)}$$
$$= \frac{13466}{627 \cdot 2 - 89 \cdot 6} = \text{say, } 25 \text{ feet}$$

The influence of a rising gradient being all in favour of the brakes need not be taken into account in calculating the strength of the rods, etc., if these are made strong enough to take the strains when stopping the car on the level or going downhill. Fig. 40 shows gradients of various degrees expressed as percentages.

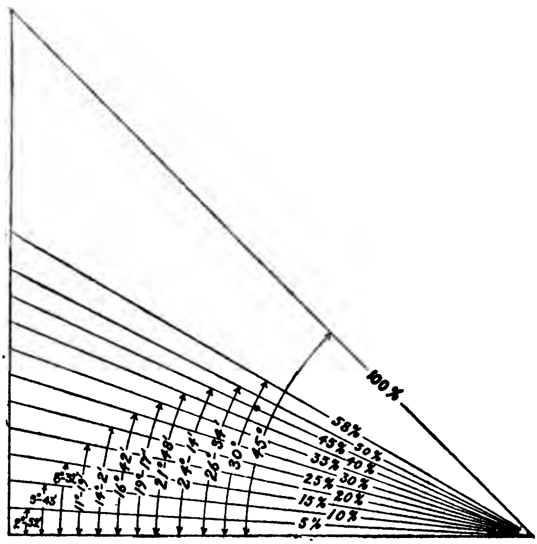


FIG. 40.

A properly designed band brake should be equally effective whether the car is running forward or backward, and the rods, etc., on both ends of the brake band should therefore be strong enough to stand the pull p.

To provide for cases where the brakes have abnormal strains to bear, such as when the road wheels are skidded, the pull p found by formula 58 should be doubled. In our example this will give a pull of 1890 lbs. on each brake band. Using good quality mild steel, with a

tenacity of 80,000 lbs. per square inch, and allowing a factor of safety of 5, the sectional area of the brake band or the actuating rod at the weakest place should not be less than—

$$\frac{1890 \times 5}{80000} = 0{\cdot}118 \text{ square inch}$$

or, say, a diameter of $\frac{7}{16}$ inch at the bottom of the threads on the pull rods. When the brake bands are of the type known as single-acting, as in Fig. 41, the pull on that end of the brake band which is acted upon by the operator (marked p in Fig. 41) will be less than the pull p as found

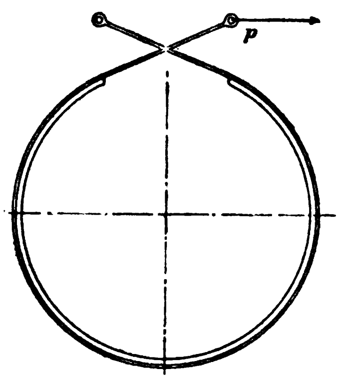


FIG. 41.

above, owing to the winding action of the drum tending to tighten the band. As the amount of surface encircled by the brake band increases, the pull p_1 will be lessened. If we take c = the coefficient of friction between the brake drum and the band, and θ = the number of degrees in the angle encircled by the brake band, we have—

$$\text{(60)} \qquad \text{Log}\,\frac{p}{p_1} = 0{\cdot}434c\theta$$

Tabulating values for various angles, from this we obtain—

TABLE 15.

Degrees. θ	Part of circumference.	$\frac{p}{p_1}$	p being = 1, p_1 =
180	0·5	3·5	0·28
240	0·666	5·34	0·19
270	0·75	6·6	0·15
300	0·833	8·1	0·12
315	0·875	9·0	0·11
360	1·0	12·85	0·08

A truly double-acting band brake will retard the vehicle equally well running either forward or backward,

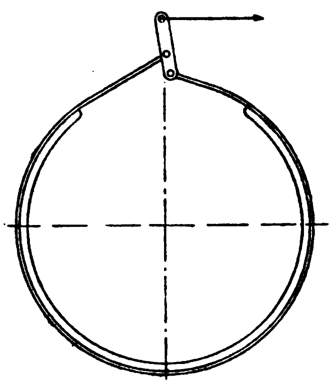


FIG. 42.

and Fig. 42 is an example of such a brake. Modern practice inclines to the use of "calliper" brakes, in which all the frictional surfaces are of metal. Brake bands with leather or similar substances as linings are unreliable. By continuous use, as when descending long inclines, the

lining is apt to char, owing to the heat generated by the friction. Metallic brakes cannot burn, but nevertheless on heavy cars the brake drum should be cooled by water circulation. The writer favours the design of calliper brake seen in Fig. 43. In this the pull on the two brake shoes is equalized by the system of levers adopted. The shoes are pivoted at A to a lug provided on the axle.

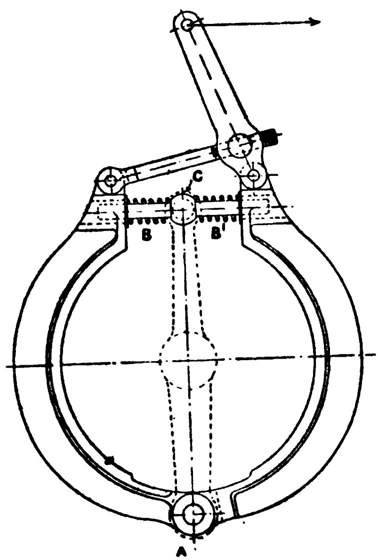


FIG. 43.

The springs B, B′ keep the shoes out of contact with the drum when the brake is not in action, the arm C serving as an abutment for the springs, and to keep the shoes concentric with the drum.

It is important that the two road wheels should be equally retarded when the brakes are applied, especially when the road surface is at all slippery. Unequal braking effort is a potent cause of "side slip," and, even when there

is no danger of slipping, causes unnecessary wear of the tyres and unequal strains in the vehicle. There seems to be no other reason than slightly increased cost against the use of spur, or bevel, differential gearing for equalizing the pull on the two brake rods. The writer does not consider it good practice to transmit the actual braking effort through the medium of the balance gear, for should the frictional resistance between the tyre and the road be less for one wheel than the other, side slip is invited. The brakes must retard each wheel equally and independently of the balance gear. The use of balance gearing to equalize the pull on the brake rods is not affected by this consideration.

BALL BEARINGS.

Ball bearings find many applications in an automobile as thrust bearings and shaft bearings. When well proportioned for their load and the speed of the shaft, they give great freedom from friction with a minimum of lubrication and adjustment. Ball bearings are more suited to slow than high speeds, especially when heavy pressures are in question, and are more useful as thrust bearings than in any other capacity, such as behind bevel-gear wheels or worm gears, where the load is steady. Where shocks are to be encountered, ball bearings have no place, owing to the liability of the balls to split.

Thrust bearings are usually of the four-point type, an illustration of which is seen in Fig. 44. To design a bearing of this kind, the number and diameter of the balls must first be decided upon from consideration of the load and speed. The radius of the pitch-circle of the balls can then be determined from—

$$(61) \qquad R = \frac{r}{\sin \frac{1}{2}\theta}$$

where R = the radius of the pitch-circle, r = the radius of one ball, and θ = the angle subtended by the ball, as in Fig. 44.

The angle θ can be obtained from—

$$(62) \qquad \theta = \frac{360°}{N}$$

where N = the number of balls it is proposed to use. Table No. 46, p. 170, will be of service in this connection.

The points A and B may be located anywhere on the vertical line CD. The circle shown passing through the centres of the balls is the pitch-circle. The bearing surfaces of the ball races are described by drawing lines from the points A and B tangent to the circles representing

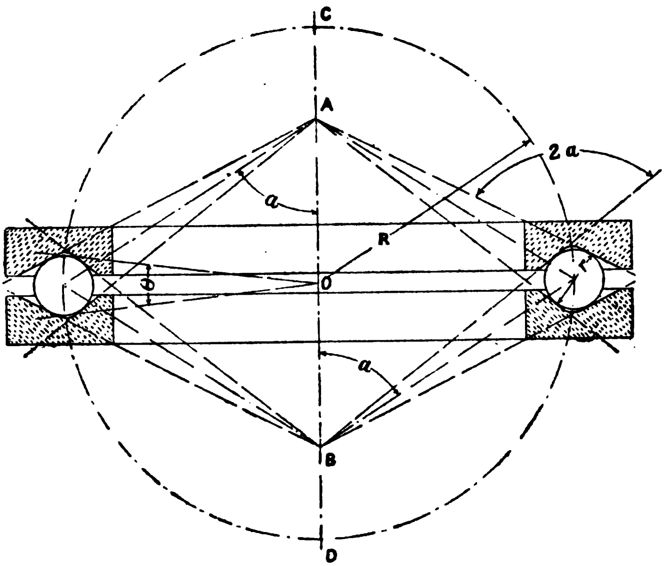


FIG. 44.

the two balls. If, instead of drawing the tangents from A and B, the points C and D at the intersections of the pitch-circle with the vertical line are used, the angles *a*, *a* will each be 45°.

To assist in determining the size of ball to use, the following table will be useful:—

TABLE 16.

Diameter of ball.	Crushing load.	Working load.	Weight per gross.
inch	lbs.	lbs.	lbs.
$\frac{1}{8}$	1,288	160	0·0415
$\frac{3}{16}$	2,900	360	0·1401
$\frac{1}{4}$	5,150	640	0·3322
$\frac{5}{16}$	8,050	1000	0·6488
$\frac{3}{8}$	11,600	1450	1·1213
$\frac{1}{2}$	20,600	2570	2·6576

In designing a ball bearing, whether for taking a thrust or carrying a shaft, a certain amount of clearance between the balls is necessary to allow them freedom of movement, and this should be about 0·005 inch between adjacent balls, but the total amount allowed in any bearing should not exceed one-third the diameter of the ball used.

From formula 61 the table No. 17, below, has been calculated, and will be found to save time when setting out a bearing.

TABLE 17.

DIAMETER OF BALL PITCH-CIRCLES, CALCULATED FOR BALLS OF 1-INCH DIAMETER.

Number of balls.	Diameter of pitch-circle.	Number of balls.	Diameter of pitch-circle.
	ins.		ins.
6	2·000	18	5·758
7	2·310	19	6·075
8	2·612	20	6·394
9	2·923	21	6·710
10	3·236	22	7·027
11	3·548	23	7·345
12	3·864	24	7·662
13	4·179	25	7·978
14	4·494	26	8·296
15	4·810	27	8·615
16	5·125	28	8·934
17	5·440	30	9·566

It may be assumed that the friction of a ball bearing is independent of the speed and the number of balls, but the ball used should be as large as possible, as the friction varies, roughly speaking, as the square of the diameter of the ball in inverse ratio. To provide a good factor of safety, and to allow for unequal distribution of the load, it is usual to assume that the whole load is carried by one ball, and to design accordingly.

The safe loads given in the above table are calculated for a speed of 150 revolutions per minute of the bearing. As the speed is increased the value of the safe load will be decreased, it being safe to assume that, should the speed be doubled, the load should be decreased by one-third. To determine the pitch-circle diameter for balls of any other diameter than 1 inch, multiply the tabular number in Table 17 by the diameter of the ball selected.

CARRIAGE SPRINGS.

THE available data on the design of carriage springs is very limited, the method most often followed being a process of trial and error. For single springs, known as "grasshopper" springs, the writer uses the formula given by D. K. Clarke, as follows—

$$(63) \qquad \text{Safe load in tons} = \frac{BT^2N}{CS}$$

in which B = width of plates in inches.
T = thickness of plates in $\frac{1}{16}$ inch.
N = number of plates in spring.
S = span of spring in inches.
C = constant = 11·3.

To determine the deflection in inches per ton of load, the most reliable formula the writer is acquainted with is that given in the *Practical Engineer* pocket-book, and repeated here—

$$(64) \qquad D = \frac{L^3}{CBT^3N}$$

where D = deflection in inches per ton of load.
L = span of spring in inches.
C = constant = 40,000 for single and 20,000 for double springs.
B = width of plates in inches.

T = thickness of plates in inches.
N = number of plates in the spring.

In a good many instances it will be found that the pneumatic tyres have more to do with the comfort of the occupants of the car than the springs, at least in the case of cars built a short time back. The present tendency to use long springs is a step in the right direction as tending to easier riding, which not only adds to the comfort of riding, but increases the life of the mechanism by protecting it from shock and vibration.

The springs of a car should be of such a strength that they will be deflected through half their working distance when fully loaded. When first put to work it will probably be found that the springs will take a permanent set, so that in estimating the initial deflection to be allowed this should receive consideration. The amount of this set will only be slight, and will vary somewhat with different springs, according to the tempering.

It is recommended that copious notes be taken, whenever possible, of springs in actual use, and these, when tabulated, will be of more practical value than any formulæ.

When unloaded with either passengers or fuel, the car frame should not be parallel with the ground line, but should be slightly higher at the back than in front. When the load is taken on board, the back end of the frame will be depressed, and should then be parallel with the ground. If the back of the frame is allowed to be lower than the front, *i.e.* nearer the ground, the appearance of the car will suffer. If anything, it will be better to let the front of the frame be lower than the back.

APPENDIX.

TABLE 18.

Areas of Small Circles.

Diameter.	Areas.									
	·00	·01	·02	·03	·04	·05	·06	·07	·08	·09
·0	·0	·000078	·00031	·0007	·00125	·00196	·00283	·00385	·00503	·00636
·1	·0078	·0095	·0113	·0133	·0154	·0177	·0201	·0227	·0255	·0283
·2	·0314	·03464	·038	·0415	·0452	·0491	·0531	·0572	·0616	·066
·3	·0706	·0755	·0804	·0855	·0908	·0962	·1018	·1075	·1134	·1195
·4	·1256	·132	·1385	·1452	·1520	·1590	·1662	·1735	·181	·1886
·5	·1963	·2043	·2124	·2206	·2290	·2376	·2463	·2552	·2642	·2734
·6	·2827	·2922	·3014	·3117	·3217	·3318	·3421	·3526	·3632	·3739
·7	·3848	·3959	·4071	·4185	·4301	·4418	·4536	·4657	·4778	·4902
·8	·5026	·5153	·5281	·5411	·5542	·5674	·5809	·5945	·6082	·6221
·9	·6362	·6504	·6648	·6793	·694	·7088	·7238	·739	·7543	·7698

TABLE 19.

Useful Functions of π.

$\pi = 3{\cdot}14159265$	$\frac{\pi}{7} = 0{\cdot}44879895$	$\pi^2 = 9{\cdot}86960440$	$\sqrt{\pi} = 1{\cdot}77245385$
$\frac{\pi}{2} = 1{\cdot}57079633$	$\frac{\pi}{16} = 0{\cdot}19634954$	$\pi^3 = 31{\cdot}00627668$	$\sqrt[3]{\pi} = 1{\cdot}46459189$
$\frac{\pi}{3} = 1{\cdot}04719755$	$\frac{\pi}{24} = 0{\cdot}13089969$	$\frac{1}{\pi} = 0{\cdot}31830989$	$\frac{1}{\sqrt{\pi}} = 0{\cdot}56418958$
$\frac{\pi}{4} = 0{\cdot}78539816$	$\frac{\pi}{32} = 0{\cdot}09817477$	$\frac{1}{\pi^2} = 0{\cdot}10132118$	$\frac{1}{\sqrt[3]{\pi}} = 0{\cdot}68278406$
$\frac{\pi}{6} = 0{\cdot}52359878$	$\frac{\pi}{180} = 0{\cdot}01745329$	$\frac{1}{\pi^3} = 0{\cdot}03225153$	$\text{Log}\ \pi = 0{\cdot}49714987$

TABLE 20.

Areas of Small Circles, advancing by Decimals.

Diameter.	Area.									
	·000	·001	·002	·003	·004	·005	·006	·007	·008	·009
·000	0	·0000008	·0000031	·0000071	·0000126	·0000196	·0000283	·0000385	·0000503	·0000636
·010	·0000785	·0000950	·0001131	·0001327	·0001539	·0001767	·0002016	·0002270	·0002565	·0002835
·020	·0003142	·0003464	·0003801	·0004155	·0004524	·0004909	·0005309	·0005726	·0006158	·0006605
·030	·0007069	·0007548	·0008043	·0008553	·0009079	·0009621	·0010179	·0010752	·0011341	·0011946
·040	·0012566	·0013203	·0013854	·0014522	·0015205	·0015904	·0016619	·0017349	·0018006	·0018857
·050	·0019635	·0020428	·0021237	·0022062	·0022902	·0023758	·0024630	·0025517	·0026421	·0027340
·060	·0028274	·0029225	·0030191	·0031172	·0032170	·0033183	·0034212	·0035257	·0036317	·0037393
·070	·0038484	·0039592	·0040715	·0041854	·0043008	·0044179	·0045365	·0046566	·0047784	·0049017
·080	·0050276	·0051530	·0052810	·0054106	·0055418	·0056745	·0058088	·0059447	·0060821	·0062211
·090	·0063617	·0065039	·0066476	·0067929	·0069398	·0070882	·0072382	·0073898	·0075430	·0076977

TABLE 21.

Areas of Circles up to 6 Inches Diameter, advancing by 32nds and 16ths.

Diameter.	Circumference.	Area.	Diameter.	Circumference.	Area.	Diameter.	Circumference.	Area.
$\frac{1}{32}$	·0981	·00077	$1\frac{3}{8}$	4·3197	1·4848	$3\frac{11}{16}$	11·584	10·679
$\frac{1}{16}$	·1963	·00307	$1\frac{7}{16}$	4·516	1·6229	$3\frac{3}{4}$	11·781	11·044
$\frac{3}{32}$	·2945	·0069	$1\frac{1}{2}$	4·7124	1·7671	$3\frac{13}{16}$	11·977	11·416
$\frac{1}{8}$	·3927	·01227	$1\frac{9}{16}$	4·9087	1·9175	$3\frac{7}{8}$	12·173	11·793
$\frac{5}{32}$	·4908	·0192	$1\frac{5}{8}$	5·1051	2·0739	$3\frac{15}{16}$	12·369	12·177
$\frac{3}{16}$	·589	·02761	$1\frac{11}{16}$	5·3014	2·2365	4	12·566	12·566
$\frac{7}{32}$	·6872	·0376	$1\frac{3}{4}$	5·4978	2·4052	$4\frac{1}{16}$	12·762	12·962
$\frac{1}{4}$	·7854	·04909	$1\frac{13}{16}$	5·6941	2·58	$4\frac{1}{8}$	12·959	13·364
$\frac{9}{32}$	·8835	·0621	$1\frac{7}{8}$	5·8905	2·7611	$4\frac{3}{16}$	13·155	13·772
$\frac{5}{16}$	·9817	·0767	$1\frac{15}{16}$	6·0868	2·9483	$4\frac{1}{4}$	13·351	14·186
$\frac{11}{32}$	1·0799	·0928	2	6·2832	3·1416	$4\frac{5}{16}$	13·547	14·606
$\frac{3}{8}$	1·1781	·1104	$2\frac{1}{16}$	6·4795	3·3410	$4\frac{3}{8}$	13·744	15·033
$\frac{13}{32}$	1·2762	·1296	$2\frac{1}{8}$	6·6759	3·5465	$4\frac{7}{16}$	13·94	15·465
$\frac{7}{16}$	1·3744	·1503	$2\frac{3}{16}$	6·8722	3·7584	$4\frac{1}{2}$	14·137	15·904
$\frac{15}{32}$	1·4726	·1725	$2\frac{1}{4}$	7·0686	3·976	$4\frac{9}{16}$	14·333	16·394
$\frac{1}{2}$	1·5708	·1963	$2\frac{5}{16}$	7·2649	4·2	$4\frac{5}{8}$	14·529	16·8
$\frac{17}{32}$	1·6689	·2216	$2\frac{3}{8}$	7·4613	4·4302	$4\frac{11}{16}$	14·725	17·257
$\frac{9}{16}$	1·7771	·2485	$2\frac{7}{16}$	7·6576	4·6664	$4\frac{3}{4}$	14·922	17·72
$\frac{19}{32}$	1·8653	·2768	$2\frac{1}{2}$	7·854	4·9087	$4\frac{13}{16}$	15·119	18·19
$\frac{5}{8}$	1·9635	·3068	$2\frac{9}{16}$	8·0503	5·1573	$4\frac{7}{8}$	15·315	18·665
$\frac{21}{32}$	2·0616	·3382	$2\frac{5}{8}$	8·2467	5·4119	$4\frac{15}{16}$	15·511	19·147
$\frac{11}{16}$	2·1598	·3712	$2\frac{11}{16}$	8·443	5·6723	5	15·708	19·635
$\frac{23}{32}$	2·258	·4057	$2\frac{3}{4}$	8·6394	5·9395	$5\frac{1}{16}$	15·904	20·129
$\frac{3}{4}$	2·3562	·4417	$2\frac{13}{16}$	8·8357	6·2126	$5\frac{1}{8}$	16·1	20·629
$\frac{25}{32}$	2·4543	·4793	$2\frac{7}{8}$	9·0321	6·4918	$5\frac{3}{16}$	16·296	21·135
$\frac{13}{16}$	2·5525	·5185	$2\frac{15}{16}$	9·2284	6·7772	$5\frac{1}{4}$	16·493	21·647
$\frac{27}{32}$	2·6507	·5591	3	9·4248	7·0686	$5\frac{5}{16}$	16·689	21·166
$\frac{7}{8}$	2·7489	·6013	$3\frac{1}{16}$	9·6211	7·3662	$5\frac{3}{8}$	16·886	22·69
$\frac{29}{32}$	2·847	·645	$3\frac{1}{8}$	9·8175	7·6699	$5\frac{7}{16}$	17·082	23·221
$\frac{15}{16}$	2·9452	·6903	$3\frac{3}{16}$	10·014	7·9798	$5\frac{1}{2}$	17·278	23·758
$\frac{31}{32}$	3·0434	·737	$3\frac{1}{4}$	10·21	8·2957	$5\frac{9}{16}$	17·474	24·301
1	3·1416	·7854	$3\frac{5}{16}$	10·406	8·618	$5\frac{5}{8}$	17·671	24·85
$1\frac{1}{16}$	3·3379	·8866	$3\frac{3}{8}$	10·602	8·9462	$5\frac{11}{16}$	17·867	25·406
$1\frac{1}{8}$	3·5343	·994	$3\frac{7}{16}$	10·799	9·2807	$5\frac{3}{4}$	18·064	25·967
$1\frac{3}{16}$	3·7306	1·1075	$3\frac{1}{2}$	10·995	9·6211	$5\frac{13}{16}$	18·261	26·535
$1\frac{1}{4}$	3·927	1·2271	$3\frac{9}{16}$	11·191	9·968	$5\frac{7}{8}$	18·457	27·108
$1\frac{5}{16}$	4·1233	1·353	$3\frac{5}{8}$	11·388	10·32	$5\frac{15}{16}$	18·653	27·688

TABLE 22.

Areas of Circles, advancing by 10ths.

Diameter.	Areas. ·0	·1	·2	·3	·4	·5	·6	·7	·8	·9	Diameter.
0	·0	·0078	·0314	·0706	·1256	·1963	·2827	·3848	·5026	·6361	0
1	·7854	·9503	1·1309	1·3273	1·5393	1·7671	2·0106	2·2698	2·5446	2·8352	1
2	3·1416	3·4636	3·8013	4·1547	4·5239	4·9087	5·3093	5·7255	6·1575	6·6052	2
3	7·0686	7·5476	8·0424	8·5530	9·0792	9·6211	10·1787	10·7521	11·3411	11·9459	3
4	12·5664	13·2025	13·8544	14·5220	15·2053	15·9043	16·6190	17·3494	18·0951	18·8574	4
5	19·6350	20·4282	21·2372	22·0618	22·9022	23·7583	24·6301	25·5176	26·4208	27·3397	5
6	28·2744	29·2247	30·1907	31·1725	32·1699	33·1831	34·2120	35·2566	36·3168	37·3928	6
7	38·4846	39·5920	40·7151	41·8539	43·0085	44·1787	45·3647	46·5663	47·7837	49·0168	7
8	50·2656	51·5300	52·8102	54·1062	55·4178	56·5471	58·0881	59·4469	60·8213	62·2115	8
9	63·6174	65·0389	66·4762	67·9292	69·3979	70·8823	72·3824	73·8982	75·4298	76·9770	9
10	78·5400	80·1186	81·7130	83·3230	84·9488	86·5903	88·2475	89·9204	91·6090	93·3133	10
11	95·0334	96·7691	98·5205	100·287	102·070	103·869	105·683	107·513	109·359	111·220	11
12	113·097	114·990	116·898	118·823	120·763	122·718	124·690	126·677	128·679	130·698	12
13	132·732	134·782	136·848	138·929	141·026	143·139	145·267	147·411	149·571	151·747	13
14	153·938	156·145	158·368	160·606	162·860	165·130	167·415	169·717	172·034	174·366	14
15	176·715	179·079	181·458	183·854	186·265	188·692	191·134	193·593	196·067	198·556	15
16	201·062	203·583	206·120	208·672	211·241	213·825	216·424	219·040	221·671	224·318	16
17	226·980	229·658	232·352	235·062	237·787	240·528	243·285	246·057	248·846	251·650	17
18	254·469	257·304	260·125	263·022	265·905	268·803	271·716	274·646	277·591	280·552	18
19	283·529	286·521	289·529	292·553	295·593	298·648	301·719	304·805	307·908	311·026	19
20	314·160	317·309	320·474	323·655	326·852	330·064	333·292	336·536	339·795	343·070	20
21	346·361	349·667	352·990	356·328	359·681	363·051	366·436	369·837	373·253	376·685	21
22	380·133	383·597	387·076	390·571	394·082	397·608	401·150	404·708	408·282	411·871	22
23	415·476	419·097	422·733	426·385	430·053	433·737	437·436	441·151	444·881	448·628	23
24	452·390	456·168	459·961	463·770	467·595	471·436	475·292	479·164	483·052	486·955	24
25	490·875	494·809	498·760	502·726	506·708	510·706	514·719	518·748	522·793	526·854	25

TABLE 23.

Circumferences of Circles, advancing by 8ths.

Diam.	Circumferences.								Diam.
	·0	1/8	1/4	3/8	1/2	5/8	3/4	7/8	
0	·0	·392	·735	1·178	1·57	1·963	2·356	2·748	0
1	3·141	3·534	3·927	4·319	4·712	5·105	5·497	5·89	1
2	6·283	6·675	7·068	7·461	7·854	8·246	8·639	9·032	2
3	9·424	9·817	10·21	10·602	10·995	11·388	11·781	12·173	3
4	12·566	12·959	13·351	13·744	14·137	14·529	14·922	15·315	4
5	15·708	16·1	16·493	16·886	17·278	17·671	18·064	18·456	5
6	18·849	19·242	19·635	20·027	20·42	20·813	21·205	21·598	6
7	21·991	22·383	22·776	23·169	23·562	23·954	24·347	24·74	7
8	25·132	25·525	25·918	26·31	26·703	27·096	27·489	27·881	8
9	28·274	28·667	29·059	29·452	29·845	30·237	30·63	31·023	9
10	31·416	31·808	32·201	32·594	32·986	33·379	33·772	34·164	10
11	34·557	34·95	35·343	35·735	36·128	36·521	36·913	37·306	11
12	37·699	38·091	38·484	38·877	39·27	39·662	40·055	40·448	12
13	40·84	41·233	41·626	42·018	42·411	42·804	43·197	43·589	13
14	43·982	44·375	44·767	45·16	45·553	45·945	46·338	46·731	14
15	47·124	47·516	47·909	48·302	48·694	49·087	49·48	49·873	15
16	50·265	50·658	51·051	51·443	51·836	52·229	52·621	53·014	16
17	53·407	53·799	54·192	54·585	54·978	55·37	55·763	56·156	17
18	56·548	56·941	57·334	57·726	58·119	58·512	58·905	59·297	18
19	59·69	60·083	60·475	60·868	61·261	61·653	62·046	62·439	19
20	62·832	63·224	63·617	64·01	64·402	64·795	65·188	65·58	20
21	65·973	66·366	66·759	67·151	67·544	67·937	68·329	68·722	21
22	69·115	69·507	69·9	70·293	70·686	71·078	71·471	71·864	22
23	72·256	72·649	73·042	73·434	73·827	74·22	74·613	75·005	23
24	75·398	75·791	76·183	76·576	76·969	77·361	77·754	78·147	24
25	78·54	78·932	79·325	79·718	80·11	80·503	80·896	81·288	25
26	81·681	82·074	82·467	82·859	83·252	83·645	84·037	84·43	26
27	84·823	85·215	85·608	86·001	86·394	86·786	87·179	87·572	27
28	87·964	88·357	88·75	89·142	89·535	89·928	90·321	90·713	28
29	91·106	91·499	91·891	92·284	92·677	93·069	93·462	93·855	29
30	94·248	94·64	95·033	95·426	95·818	96·211	96·604	96·996	30
31	97·389	97·782	98·175	98·567	98·96	99·353	99·745	100·138	31
32	100·531	100·923	101·316	101·709	102·102	102·494	102·887	103·28	32
33	103·672	104·065	104·458	104·85	105·243	105·636	106·029	106·421	33
34	106·814	107·207	107·6	107·992	108·385	108·778	109·171	109·563	34
35	109·956	110·349	110·741	111·134	111·527	111·919	112·312	112·705	35
36	113·098	113·49	113·883	114·276	114·668	115·061	115·454	115·846	36
37	116·239	116·632	117·025	117·417	117·81	118·203	118·595	118·988	37
38	119·381	119·773	120·166	120·559	120·952	121·344	121·737	122·130	38
39	122·522	122·915	123·308	123·7	124·093	124·486	124·879	125·271	39
40	125·664	126·057	126·449	126·842	127·235	127·627	128·02	128·413	40
41	128·806	129·198	129·591	129·984	130·376	130·769	131·162	131·554	41
42	131·947	132·34	132·733	133·125	133·518	133·911	134·303	134·696	42
43	135·089	135·481	135·874	136·267	136·66	137·052	137·445	137·838	43
44	138·23	138·623	139·016	139·408	139·801	140·194	140·587	140·979	44
45	141·372	141·765	142·157	142·55	142·943	143·335	143·728	144·121	45
46	144·514	144·906	145·299	145·692	146·084	146·477	146·87	147·262	46
47	147·655	148·048	148·441	148·833	149·226	149·619	150·011	150·404	47
48	150·797	151·189	151·582	151·975	152·368	152·76	153·153	153·545	48

TABLE 24.

Circumferences of Circles, advancing by 10ths.

Diam.	·0	·1	·2	·3	·4	·5	·6	·7	·8	·9	Diam.
0	·00	·31	·62	·94	1·25	1·57	1·88	2·19	2·51	2·82	0
1	3·14	3·45	3·77	4·08	4·39	4·71	5·02	5·34	5·65	5·96	1
2	6·28	6·59	6·91	7·22	7·53	7·85	8·16	8·48	8·79	9·11	2
3	9·42	9·74	10·05	10·36	10·68	10·99	11·30	11·62	11·93	12·25	3
4	12·56	12·88	13·19	13·50	13·82	14·13	14·45	14·76	15·08	15·39	4
5	15·70	16·02	16·33	16·65	16·96	17·27	17·59	17·90	18·22	18·53	5
6	18·84	19·16	19·47	19·79	20·10	20·42	20·73	21·04	21·36	21·67	6
7	21·99	22·30	22·61	22·93	23·24	23·56	23·87	24·19	24·50	24·81	7
8	25·13	25·44	25·76	26·07	26·38	26·70	27·01	27·33	27·64	27·96	8
9	28·27	28·58	28·90	29·21	29·53	29·84	30·15	30·47	30·78	31·10	9
10	31·41	31·73	32·04	32·35	32·67	32·98	33·30	33·61	33·92	34·24	10
11	34·55	34·87	35·18	35·50	35·81	36·12	36·44	36·75	37·07	37·38	11
12	37·69	38·01	38·32	38·64	38·95	39·27	39·58	39·89	40·21	40·52	12
13	40·84	41·15	41·46	41·78	42·09	42·41	42·72	43·03	43·35	43·66	13
14	43·98	44·29	44·61	44·92	45·23	45·55	45·86	46·18	46·49	46·80	14
15	47·12	47·43	47·75	48·06	48·38	48·69	49·00	49·32	49·63	49·95	15
16	50·26	50·57	50·89	51·20	51·52	51·83	52·15	52·46	52·78	53·09	16
17	53·40	53·72	54·03	54·35	54·65	54·97	55·29	55·60	55·92	56·23	17
18	56·54	56·86	57·17	57·49	57·80	58·11	58·43	58·74	59·06	59·37	18
19	59·69	60·00	60·31	60·63	60·94	61·26	61·57	61·88	62·20	62·51	19
20	62·83	63·14	63·46	63·77	64·08	64·40	64·71	65·03	65·34	65·65	20
21	65·97	66·28	66·60	66·91	67·22	67·54	67·85	68·17	68·48	68·80	21
22	69·11	69·42	69·74	70·05	70·37	70·68	71·00	71·31	71·62	71·94	22
23	72·25	72·57	72·88	73·19	73·51	73·82	74·14	74·45	74·76	75·08	23
24	75·39	75·71	76·02	76·34	76·65	76·96	77·28	77·59	77·91	78·22	24
25	78·54	78·85	79·16	79·48	79·79	80·11	80·42	80·73	81·05	81·36	25
26	81·68	81·99	82·30	82·62	82·93	83·25	83·56	83·88	84·19	84·50	26
27	84·82	85·13	85·45	85·76	86·07	86·39	86·70	87·02	87·33	87·65	27
28	87·96	88·27	88·59	88·90	89·22	89·53	89·84	90·16	90·47	90·79	28
29	91·10	91·42	91·73	92·04	92·36	92·67	92·99	93·30	93·61	93·93	29
30	94·24	94·56	94·87	95·19	95·50	95·81	96·13	96·44	96·76	97·07	30
31	97·38	97·70	98·01	98·33	98·64	98·96	99·27	99·58	99·90	100·2	31
32	100·5	100·8	101·1	101·4	101·7	102·1	102·4	102·7	103·0	103·3	32
33	103·6	103·9	104·3	104·6	104·9	105·2	105·5	105·8	106·1	106·5	33
34	106·8	107·1	107·4	107·7	108·0	108·3	108·6	109·0	109·3	109·6	34
35	109·9	110·2	110·5	110·8	111·2	111·5	111·8	112·1	112·4	112·7	35
36	113·0	113·4	113·7	114·0	114·3	114·6	114·9	115·2	115·6	115·9	36
37	116·2	116·5	116·8	117·1	117·4	117·8	118·1	118·4	118·7	119·0	37
38	119·3	119·6	120·0	120·3	120·6	120·9	121·2	121·5	121·8	122·2	38
39	122·5	122·8	123·1	123·4	123·7	124·0	124·4	124·7	125·0	125·3	39
40	125·6	125·9	126·2	126·6	126·9	127·2	127·5	127·8	128·1	128·4	40
41	128·8	129·1	129·4	129·7	130·0	130·3	130·6	131·0	131·3	131·6	41
42	131·9	132·2	132·5	132·8	133·2	133·5	133·8	134·1	134·4	134·7	42
43	135·0	135·4	135·7	136·0	136·3	136·6	136·9	137·2	137·6	137·9	43
44	138·2	138·5	138·8	139·1	139·4	139·8	140·1	140·4	140·7	141·0	44
45	141·3	141·6	142·0	142·3	142·6	142·9	143·2	143·5	143·9	144·2	45
46	144·5	144·8	145·1	145·4	145·7	146·0	146·3	146·7	147·0	147·3	46
47	147·6	147·9	148·2	148·5	148·9	149·2	149·5	149·8	150·1	150·4	47
48	150·7	151·1	151·4	151·7	152·0	152·3	152·6	152·9	153·3	153·6	48

TABLE 25.

Table of Squares, Cubes, etc.

No.	Square.	Cube.	Square root.	Cube root.	No.	Square.	Cube.	Square root.	Cube root.
1	1	1	1	1	5	25	125	2·2361	1·71
1·1	1·21	1·331	1·0488	1·0323	5·1	26·01	132·651	2·2583	1·7213
1·2	1·44	1·728	1·0955	1·0627	5·2	27·04	140·608	2·2804	1·7325
1·3	1·69	2·197	1·1402	1·0914	5·3	28·09	148·877	2·3022	1·7435
1·4	1·96	2·744	1·1832	1·1187	5·4	29·16	157·464	2·3238	1·7544
1·5	2·25	3·375	1·2247	1·1447	5·5	30·25	166·375	2·3452	1·7652
1·6	2·56	4·096	1·2649	1·1696	5·6	31·36	175·616	2·2664	1·7758
1·7	2·89	4·913	1·3038	1·1935	5·7	32·49	185·193	2·3875	1·7863
1·8	3·24	5·832	1·3416	1·2164	5·8	33·64	195·112	2·4083	1·7967
1·9	3·61	6·859	1·3784	1·2386	5·9	34·81	205·379	2·429	1·807
2	4	8	1·4142	1·2599	6	36	216	2·4495	1·8171
2·1	4·41	9·261	1·4491	1·2806	6·1	37·21	226·981	2·4698	1·8272
2·2	4·84	10·648	1·4832	1·3006	6·2	38·44	238·328	2·49	1·8371
2·3	5·29	12·167	1·5166	1·32	6·3	39·69	250·047	2·51	1·8469
2·4	5·76	13·824	1·5492	1·3389	6·4	40·96	262·144	2·5298	1·8566
2·5	6·25	15·625	1·5811	1·3572	6·5	42·25	274·625	2·5495	1·8663
2·6	6·76	17·576	1·6125	1·3751	6·6	43·56	287·496	2·569	1·8758
2·7	7·29	19·683	1·6432	1·3925	6·7	44·89	300·763	2·5884	1·8852
2·8	7·84	21·952	1·6733	1·4095	6·8	46·24	314·432	2·6077	1·8945
2·9	8·41	24·389	1·7029	1·426	6·9	47·61	328·509	2·6268	1·9038
3	9	27	1·7321	1·4422	7	49	343	2·6458	1·9129
3·1	9·61	29·791	1·7607	1·4581	7·1	50·41	357·911	2·6646	1·922
3·2	10·24	32·768	1·7889	1·4736	7·2	51·84	373·248	2·6833	1·931
3·3	10·89	35·937	1·8166	1·4888	7·3	53·29	389·017	2·7019	1·9399
3·4	11·56	39·304	1·8439	1·5037	7·4	54·76	405·224	2·7203	1·9487
3·5	12·25	42·875	1·8708	1·5183	7·5	56·25	421·875	2·7386	1·9574
3·6	12·96	46·656	1·8974	1·5326	7·6	57·76	438·976	2·7568	1·9661
3·7	13·69	50·653	1·9235	1·5467	7·7	59·29	456·533	2·7749	1·9747
3·8	14·44	54·872	1·9494	1·5605	7·8	60·84	474·552	2·7928	1·9832
3·9	15·21	59·319	1·9748	1·5741	7·9	62·41	493·039	2·8107	1·9916
4	16	64	2	1·5874	8	64	512	2·8234	2
4·1	16·81	68·921	2·0249	1·6005	8·1	65·61	531·441	2·846	2·0083
4·2	17·64	74·088	2·0494	1·6134	8·2	67·24	551·368	2·863	2·0165
4·3	18·49	79·507	2·0736	1·6261	8·3	68·89	571·787	2·881	2·0247
4·4	19·36	85·184	2·0976	1·6386	8·4	79·56	592·704	2·8983	2·0328
4·5	20·25	91·125	2·1213	1·651	8·5	72·25	614·125	2·9155	2·0408
4·6	21·16	97·336	2·1448	1·6631	8·6	73·96	636·056	2·9326	2·0488
4·7	22·09	103·823	2·1680	1·6751	8·7	75·69	658·503	2·9496	2·0567
4·8	23·04	110·592	2·1909	1·6869	8·8	77·44	681·472	2·9665	2·0646
4·9	24·01	117·649	2·2136	1·6985	8·9	79·21	704·969	2·9833	2·0724

TABLE OF SQUARES, CUBES, ETC.—(*continued*).

No.	Square.	Cube.	Square root.	Cube root.	No.	Square.	Cube.	Square root.	Cube root.
9	81	729	3	2·0801	13·6	184·96	2515·456	3·6878	2·387
9·1	82·81	753·571	3·0166	2·0878	13·7	187·69	2571·353	3·7013	2·3928
9·2	84·64	778·688	3·0332	2·0954	13·8	190·44	2628·072	3·7148	2·3986
9·3	86·49	804·357	3·0496	2·1029	13·9	193·21	2685·619	3·7283	2·4044
9·4	88·36	830·584	3·0659	2·1105	14	196	2744	3·7417	2·4101
9·5	90·25	857·375	3·0822	2·1179	14·1	198·81	2803·221	3·755	2·4159
9·6	92·16	884·736	3·0984	2·1253	14·2	201·64	2863·288	3·7683	2·4216
9·7	94·09	912·673	3·1145	2·1327	14·3	204·49	2924·207	3·7815	2·4329
9·8	96·04	941·192	3·1305	2·14	14·4	207·36	2985·984	3·2747	2·4372
9·9	98·01	1970·299	3·1464	2·1472	14·5	210·25	3048·625	3·8079	2·4385
10	100	1000	3·1623	2·1544	14·6	213·16	3112·136	3·821	2·4441
10·1	102·01	1030·301	3·178	2·1616	14·7	216·09	3176·523	3·8341	2·4497
10·2	104·04	1061·208	3·1937	2·1687	14·8	219·04	3241·792	3·8471	2·4552
10·3	106·09	1092·727	3·2094	2·1757	14·9	223·01	3307·949	3·86	2·4607
10·4	108·16	1124·863	3·2249	2·1828	15	225	3375	3·873	2·4662
10·5	110·25	1157·625	3·2404	2·1897	15·1	222·01	3442·951	3·8859	2·4717
10·6	112·36	1191·016	3·2558	2·1967	15·2	231·04	3511·808	3·8987	2·4771
10·7	114·49	1225·043	3·2711	2·2036	15·3	234·09	3581·577	3·9115	2·4825
10·8	116·64	1259·712	3·2863	2·2104	15·4	237·16	3652·264	3·9243	2·4879
10·9	118·81	1295·029	3·3015	2·2178	15·5	340·25	3723·875	3·937	2·4933
11	121	1331	3·3166	2·2239	15·6	243·36	3796·416	3·9497	2·4987
11·1	123·21	1367·631	3·3317	2·2307	15·7	246·49	3869·893	3·9623	2·504
11·2	125·44	1406·928	3·3466	2·2374	15·8	249·64	3944·312	3·9749	2·5093
11·3	127·69	1442·897	3·3615	2·2441	15·9	252·81	4019·679	3·9875	2·5146
11·4	129·96	1481·544	3·3764	2·2506	16	256	4006	4	2·5198
11·5	132·25	1520·875	3·3912	2·2572	16·1	259·21	4173·281	4·0125	2·5251
11·6	134·56	1560·896	3·4059	2·2637	16·2	262·44	4251·528	4·0249	2·5303
11·7	136·89	1601·613	3·4205	2·2702	16·3	265·69	4330·747	4·0373	2·5355
11·8	139·24	1643·032	3·4351	2·2766	16·4	268·96	4410·944	4·0497	2·5407
11·9	141·61	1685·159	3·4496	2·2831	16·5	272·25	4492·125	4·062	2·5458
12	144	1728	3·4641	2·2894	16·6	275·56	4574·296	4·0743	2·5509
12·1	146·41	1771·561	3·4785	2·2957	16·7	278·89	4657·463	4·0866	2·5561
12·2	148·84	1815·848	3·4928	2·3021	16·8	282·24	4741·632	4·0988	2·5612
12·3	151·29	1860·867	3·5071	2·3084	16·9	285·61	4826·809	4·111	2·5662
12·4	153·76	1906·624	3·5214	2·3146	17	289	4913	4·1231	2·5713
12·5	156·25	1953·125	3·5355	2·3298	17·1	292·41	5000·211	4·1352	2·5768
12·6	158·76	2000·376	3·5496	2·327	17·2	295·84	5088·448	4·1473	2·5813
12·7	161·29	2048·383	3·5637	2·3331	17·3	299·29	5177·717	4·1593	2·5863
12·8	163·84	2097·152	3·5777	2·3391	17·4	302·76	5268·025	4·1713	2·5912
12·9	166·41	2146·689	3·5917	2·3453	17·5	306·25	5359·375	4·1833	2·5962
13	169	2197	3·6056	2·3513	17·6	309·76	5451·776	4·1952	2·6012
13·1	171·61	2248·091	3·6194	2·3573	17·7	313·29	5545·233	4·2071	2·6061
13·2	174·24	2299·968	3·6332	2·3633	17·8	316·84	5639·752	4·219	2·611
13·3	176·89	2352·637	3·6469	2·3693	17·9	320·41	5735·339	4·2308	2·6159
13·4	179·56	2406·104	3·6606	2·3752	18	324	5832	4·2426	2·6207
13·5	182·25	2460·375	3·6742	2·3811	18·1	327·61	5929·741	4·2544	2·6256

TABLE OF SQUARES, CUBES, ETC.—(*continued*).

No.	Square.	Cube.	Square root.	Cube root.	No.	Square.	Cube.	Square root.	Cube root.
18·2	331·24	6028·568	4·2661	2·6304	22·8	519·84	11852·352	4·7749	2·8356
18·3	334·89	6128·487	4·2778	2·6352	22·9	524·41	12008·989	4·7854	2·8397
18·4	338·56	6229·504	4·2895	2·64	23	529	12167	4·7958	2·8438
18·5	342·25	6331·625	4·3012	2·6448	23·1	533·61	12326·391	4·8062	2·8479
18·6	345·96	6434·856	4·3128	2·6495	23·2	538·24	12487·168	4·8166	2·8521
18·7	349·69	6539·203	4·3243	2·6543	23·3	542·89	12649·337	4·827	2·8562
18·8	353·44	6644·672	4·3359	2·659	23·4	547·56	12812·904	4·8373	2·8603
18·9	357·21	6751·269	4·3474	2·6637	23·5	552·25	12977·875	4·8477	2·8643
19	361	6859	4·3589	2·6684	23·6	556·96	13144·256	4·858	2·8684
19·1	364·81	6967·871	4·3704	2·6731	23·7	561·69	13312·053	4·8683	2·8724
19·2	368·64	7077·888	4·3818	2·6777	23·8	566·44	13481·272	4·8785	2·8765
19·3	372·49	7189·057	4·3932	2·6824	23·9	571·21	13651·919	4·8888	2·8805
19·4	376·36	7301·384	4·4045	2·6870	24	576	13824	4·899	2·8845
19·5	380·25	7414·875	4·4159	2·6916	24·1	580·81	13997·521	4·9092	2·8885
19·6	384·16	7529·536	4·4272	2·6962	24·2	585·64	14172·488	4·9193	2·8925
19·7	388·09	4645·373	4·4385	2·7008	24·3	590·49	14348·907	4·9295	2·8965
19·8	392·04	7762·392	4·4497	2·7053	24·4	595·36	14526·784	4·9396	2·9004
19·9	396·01	7880·599	4·4609	2·7099	24·5	600·25	14706·125	4·9497	2·9044
20	400	8000	4·4721	2·7144	24·6	605·16	14886·936	4·9598	2·9083
20·1	404·01	8120·601	4·4833	2·7189	24·7	610·09	15069·223	4·9699	2·9123
20·2	408·04	8242·408	4·4944	2·7234	24·8	615·04	15252·992	4·9799	2·9162
20·3	412·09	8365·427	4·5055	2·7279	24·9	620·01	15438·249	4·9899	2·9201
20·4	416·16	8489·664	4·5166	2·7324	25	625	15625	5	2·9240
20·5	420·25	8615·125	4·5277	2·7368	25·1	630·01	15813·251	5·01	2·9279
20·6	424·36	8741·816	4·5387	2·7413	25·2	635·04	16003·008	5·02	2·9318
20·7	482·49	8869·743	4·5497	2·7457	25·3	640·09	16194·277	5·0299	2·9357
20·8	432·64	8998·912	4·5607	2·7502	25·4	645·16	16387·064	5·0398	2·9395
20·9	436·81	9129·329	4·5716	2·7545	25·5	650·25	16581·375	5·0498	2·9434
21	441	9261	4·5826	2·7589	25·6	655·36	16777·216	5·0596	2·9472
21·1	445·21	9393·931	4·5935	2·7633	25·7	660·49	16974·593	5·0695	2·9511
21·2	449·44	9528·128	4·6043	2·7676	25·8	665·64	17173·512	5·0794	2·9549
21·3	453·69	9663·597	4·6152	2·772	25·9	670·81	17373·979	5·0892	2·9587
21·4	457·96	9800·344	4·626	2·7763	26	676	17576	5·099	2·9625
21·5	462·25	9938·375	4·6368	2·7806	26·1	681·21	17779·581	5·1088	2·9663
21·6	466·56	10077·696	4·6476	2·7849	26·2	686·44	17984·728	5·1186	2·9701
21·7	470·89	10218·313	4·6583	2·7893	26·3	691·69	18191·447	5·1284	2·9738
21·8	475·24	10360·232	4·669	2·7935	26·4	696·96	18399·744	5·1381	2·9776
21·9	479·61	10503·459	4·6797	2·7978	26·5	702·25	18609·625	5·1478	2·9814
22	484	10648	4·6904	2·8021	26·6	707·56	18821·096	5·1575	2·9851
22·1	488·41	10793·861	4·7011	2·8063	26·7	712·89	19034·163	5·1672	2·9888
22·2	492·84	10941·048	4·7117	2·8105	26·8	718·24	19248·832	5·1769	2·9926
22·3	497·29	11089·567	4·7223	2·8147	26·9	723·61	19465·109	5·1865	2·9963
22·4	501·76	11239·424	4·7329	2·8189	27	729	19683	5·1962	3
22·5	506·25	11390·625	4·7434	2·8231	27·1	734·41	19902·511	5·2058	3·0037
22·6	510·76	11543·176	4·7539	2·8273	27·2	739·84	20123·648	5·2154	3·0074
22·7	515·29	11697·083	4·7644	2·8314	27·3	745·29	20346·417	5·2249	3·0111

TABLE OF SQUARES, CUBES, ETC.—(*continued*).

No.	Square.	Cube.	Square root.	Cube root.	No.	Square.	Cube.	Square root.	Cube root.
27·4	750·76	20570·824	5·2345	3·0147	32	1024	32768	5·6569	3·1748
27·5	756·25	20796·875	5·244	3·0184	32·1	1030·41	33076·161	5·6656	3·1781
27·6	761·76	21024·576	5·2536	3·0221	32·2	1036·84	33386·248	5·6745	3·1814
27·7	767·29	21253·933	5·2631	3·0257	32·3	1043·29	33698·267	5·6833	3·1847
27·8	772·84	21484·952	5·2726	3·0293	32·4	1049·76	34012·224	5·6921	3·188
27·9	778·41	21717·639	5·282	3·033	32·5	1056·25	34328·125	5·7008	3·1913
28	784	21952	5·2915	3·0366	32·6	1062·76	34645·976	5·7096	3·1945
28·1	789·61	22188·041	5·3509	3·0402	32·7	1069·29	34965·783	5·7183	3·1978
28·2	795·24	22425·768	5·3104	3·0438	32·8	1075·84	35287·552	5·7271	3·201
28·3	800·89	22665·187	5·3198	3·0474	32·9	1082·41	35611·289	5·7358	3·2043
28·4	806·56	22906·304	5·3292	3·051	33	1089	35937	5·7446	3·2075
28·5	812·25	23149·125	5·3385	3·0546	33·1	1095·61	26264·691	5·7532	3·2108
28·6	817·96	23393·656	5·3479	3·0581	33·2	1102·24	36594·368	5·7619	3·214
28·7	823·69	23639·903	5·3572	3·0617	33·3	1108·89	36926·037	5·7706	3·2172
28·8	829·44	23887·872	5·3666	3·0652	33·4	1115·56	37259·704	5·7792	3·2204
28·9	835·21	24137·569	5·3759	3·0688	33·5	1122·25	37595·375	5·7879	3·2237
29	841	24389	5·3852	3·0723	33·6	1128·96	37933·056	5·7965	3·2269
29·1	846·81	24642·171	5·3944	3·0758	33·7	1135·69	38272·753	5·8051	3·2301
29·2	852·64	24897·088	5·4037	3·0794	33·8	1142·44	38614·472	5·8137	3·2332
29·3	858·49	25153·757	5·4129	3·0829	33·9	1149·21	38958·219	5·8223	3·2364
29·4	864·36	25412·184	5·4222	3·0864	34	1156	39304	5·831	3·2396
29·5	870·25	25672·375	5·4314	3·0899	34·1	1162·81	39651·821	5·8395	3·2428
29·6	876·16	25934·336	5·4406	3·0934	34·2	1169·64	40001·688	5·848	3·246
29·7	882·09	26198·073	5·4498	3·0968	34·3	1176·49	40353·607	5·8566	3·2491
29·8	888·04	26463·592	5·4589	3·1003	34·4	1183·36	40707·584	5·8651	3·2522
29·9	894·01	26730·899	5·4681	3·1038	34·5	1190·25	41063·625	5·8736	3·2554
30	900	27000	5·4772	3·1072	34·6	1197·16	41421·736	5·8821	3·2586
30·1	906·01	27270·901	5·4863	3·1107	34·7	1204·09	41781·923	5·8906	3·2617
30·2	912·04	27543·608	5·4954	3·1141	34·8	1211·04	42144·192	5·8991	3·2648
30·3	918·09	27818·127	5·5045	3·1176	34·9	1218·01	42508·549	5·9076	3·2679
30·4	924·16	28094·464	5·5136	3·121	35	1225	42875	5·9161	3·2711
30·5	930·25	28372·625	5·5226	3·1244	35·1	1232·01	43243·551	5·9245	3·2742
30·6	936·36	28652·616	5·5317	3·1278	35·2	1239·04	43614·208	5·933	3·2773
30·7	942·49	28934·443	5·5407	3·1312	35·3	1246·09	43986·977	5·9414	3·2804
30·8	948·64	29218·112	5·5497	3·1346	35·4	1253·16	44361·864	5·9498	3·2835
30·9	954·81	29503·629	5·5587	3·138	35·5	1260·25	44738·875	5·9582	3·2866
31	961	29791	5·5678	3·1414	35·6	1267·36	45118·016	5·9666	3·2897
31·1	967·21	30080·231	5·5767	3·1448	35·7	1274·49	45499·293	5·9749	3·2927
31·2	973·44	30371·328	5·5857	3·1481	35·8	1281·64	45882·712	5·9833	3·2958
31·3	979·69	30664·297	5·5946	3·1515	35·9	1288·81	46268·279	5·9917	3·2989
31·4	985·96	30959·144	5·6035	3·1548	36	1296	46656	6	3·3019
31·5	992·25	31255·875	5·6124	3·1582	36·1	1303·21	47045·881	6·0083	3·305
31·6	998·56	31554·496	5·6213	3·1615	36·2	1310·44	47437·928	6·0166	3·308
31·7	1004·89	31855·013	5·6302	3·1648	36·3	1317·69	47832·147	6·0249	3·3111
31·8	1011·24	32157·432	5·6391	3·1671	36·4	1324·96	48228·544	6·0332	3·3141
31·9	1017·61	32461·759	5·648	3·1715	36·5	1332·25	48627·125	6·0415	3·3171

TABLE OF SQUARES, CUBES, ETC.—(*continued*).

No.	Square.	Cube.	Square root.	Cube root.	No.	Square.	Cube.	Square root.	Cube root.
36·6	1339·56	49027·896	6·0498	3·3202	41·2	1697·44	69934·528	6·4187	3·4538
36·7	1346·89	49430·863	6·0581	3·3232	41·3	1705·69	70444·997	6·4265	3·4566
36·8	1354·24	49836·032	6·0663	3·3262	41·4	1713·96	70597·944	6·4343	3·4594
36·9	1361·61	50243·409	6·0745	3·3292	41·5	1722·25	71473·375	6·4421	3·4622
37	1369	50653	6·0828	3·3322	41·6	1730·56	71991·296	6·4498	3·465
37·1	1376·41	51064·811	6·091	3·3352	41·7	1738·89	72511·713	6·4575	3·4677
37·2	1383·84	51478·848	6·0992	3·3382	41·8	1747·24	73034·632	6·4653	3·4705
37·3	1391·29	51895·117	6·1074	3·3412	41·9	1755·61	73560·059	6·473	3·4733
37·4	1398·76	52313·624	6·1156	3·3442	42	1764	74088	6·4807	3·476
37·5	1406·25	52734·375	6·1237	3·3472	42·1	1772·41	74618·461	6·4884	3·4788
37·6	1413·76	53157·376	6·1319	3·3501	42·2	1780·84	75151·448	6·4961	3·4815
37·7	1421·29	53582·633	6·14	3·3531	42·3	1789·29	75686·967	6·5038	3·4843
37·8	1428·84	54010·152	6·1482	3·3561	42·4	1797·76	76225·024	6·5115	3·487
37·9	1436·41	54439·939	6·1563	3·359	42·5	1806·25	76765·625	6·5192	3·4898
38	1444	54872	6·1644	3·362	42·6	1814·76	77308·776	6·5268	3·4915
38·1	1451·61	55306·341	6·1725	3·3649	42·7	1823·29	77854·483	6·5345	3·4952
38·2	1459·24	55742·968	6·1806	3·3679	42·8	1831·84	78402·752	6·5422	3·498
38·3	1466·89	56181·887	6·1887	3·3708	42·9	1840·41	78953·589	6·5498	3·5007
38·4	1474·56	56623·104	6·1968	3·3737	43	1849	79507	6·5574	3·5034
38·5	1482·25	57066·625	6·2048	3·3767	43·1	1857·61	80062·991	6·5651	3·5061
38·6	1489·96	57512·456	6·2129	3·3796	43·2	1866·24	80621·568	6·5727	3·5088
38·7	1497·69	57960·603	6·2209	3·3825	43·3	1874·89	81182·737	6·5803	3·5115
38·8	1505·44	58411·072	6·229	3·3854	43·4	1883·56	81746·504	6·5879	3·5142
38·9	1513·21	58863·869	6·237	3·3883	43·5	1892·25	82312·875	6·5954	3·5169
39	1521	59319	6·245	3·3912	43·6	1900·96	82881·856	6·603	3·5196
39·1	1528·81	59776·471	6·253	3·3941	43·7	1909·69	83453·453	6·6106	3·5223
39·2	1536·64	60236·288	6·261	3·397	43·8	1918·44	84027·672	6·6182	3·525
39·3	1544·49	60698·457	6·269	3·3999	43·9	1927·21	84604·519	6·6257	3·5277
39·4	1552·36	61162·984	6·2769	3·4028	44	1936	85814	6·6332	3·5303
39·5	1560·25	61629·875	6·2849	3·4056	44·1	1994·81	85766·121	6·6408	3·533
39·6	1568·16	62099·136	6·2929	3·4085	44·2	1953·64	86350·888	6·6483	3·5357
39·7	1576·09	62570·773	6·3008	3·4114	44·3	1962·49	86938·307	6·6558	3·5384
39·8	1584·04	63044·792	6·3087	3·4142	44·4	1971·36	87528·384	6·6633	3·541
39·9	1592·01	63521·199	6·3166	3·4171	44·5	1980·25	88121·125	6·6708	3·5437
40	1600	64000	6·3246	3·42	44·6	1989·16	88716·536	6·6783	3·5463
40·1	1608·01	64481·201	6·3325	3·4228	44·7	1998·09	89314·623	6·6858	3·549
40·2	1616·04	64964·808	6·3404	3·4256	44·8	2007·04	89915·392	6·6933	3·5516
40·3	1624·09	65450·827	6·3482	3·4285	44·9	2016·01	90518·849	6·7007	3·5543
40·4	1632·16	65939·264	6·3561	3·4313	45	2025	91125	6·7082	3·5569
40·5	1640·25	66430·125	6·3639	3·4341	45·1	2034·01	91733·851	6·7157	3·5595
40·6	1648·36	66923·416	6·3718	3·437	45·2	2043·04	92345·408	6·7231	3·5622
40·7	1656·49	67419·143	6·3796	3·4398	45·3	2052·90	92959·677	6·7305	3·5648
40·8	1664·64	67911·312	6·3875	3·4426	45·4	2061·16	93576·664	6·738	3·5674
40·9	1672·81	68417·929	6·3953	3·4454	45·5	2070·25	94196·375	6·7454	3·57
41	1681	68921	6·4031	3·4482	45·6	2079·36	94818·816	6·7528	3·5726
41·1	1689·21	69426·531	6·4109	3·451	45·7	2088·48	95443·993	6·7602	3·5752

TABLE OF SQUARES, CUBES, ETC.—(*continued*).

No.	Square.	Cube.	Square root.	Cube root.	No.	Square.	Cube.	Square root.	Cube root.
45·8	2097·64	96071·912	6·7676	3·5778	54	2916	157464	7·3485	3·7798
45·9	2106·81	96702·579	6·775	3·5805	55	3025	166375	7·4162	3·803
46	2116	97336	6·7823	3·583	56	3136	175616	7·4833	3·8259
46·1	2125·21	97972·181	6·7897	3·5856	57	3249	185193	7·5498	3·8485
46·2	2134·44	98611·128	6·7971	3·5882	58	3364	195112	7·6158	3·8709
46·3	2143·69	99252·847	6·8044	3·5908	59	3481	205379	7·6811	3·893
46·4	2152·96	99897·344	6·8118	3·5934	60	3600	216000	7·746	3·9149
46·5	2162·25	100544·625	6·8191	3·596	61	3721	226981	7·8102	3·9365
46·6	2171·56	101194·696	6·8264	3·5986	62	3844	238328	7·874	3·9579
46·7	2180·89	101847·563	6·8337	3·6011	63	3969	250047	7·9373	3·9791
46·8	2190·24	102503·232	6·8411	3·6037	64	4096	262144	8	4
46·9	2199·61	103161·709	6·8484	3·6063	65	4225	274625	8·0623	4·0207
47	2209	103823	6·8557	3·6088	66	4356	287496	8·124	4·0412
47·1	2218·41	104487·111	6·8629	3·6114	67	4489	300768	8·1854	4·0615
47·2	2227·84	105154·048	6·8702	3·6139	68	4624	314432	8·2462	4·0817
47·3	2237·29	105823·817	6·8775	3·6165	69	4761	328509	8·3066	4·1016
47·4	2246·76	106496·424	6·8908	3·619	70	4900	343000	8·3666	4·1213
47·5	2256·25	107171·875	6·892	3·6216	71	5041	357911	8·4261	4·1408
47·6	2265·76	107850·176	6·8993	3·6241	72	5184	373248	8·4853	4·1602
47·7	2275·29	108581·333	6·9065	3·6267	73	5329	389017	8·544	4·1793
47·8	2284·84	109215·352	6·9138	3·6292	74	5476	405224	8·6023	4·1983
47·9	2294·41	109902·239	6·921	3·6317	75	5625	421875	8·6603	4·2172
48	2304	110592	6·9282	3·6342	·76	5776	438977	8·7178	4·2358
48·1	2313·61	111284·641	6·9354	3·6368	77	5929	456533	8·775	4·2543
48·2	2323·24	111930·168	6·9426	3·6393	78	6084	474552	8·8318	4·2727
48·3	2332·89	112678·587	6·9498	3·6418	79	6241	493039	8·8882	4·2908
48·4	2342·56	113379·904	6·957	3·6443	80	6400	512000	8·9443	4·3089
48·5	2352·25	114084·125	6·9642	3·6468	81	6561	531441	9	4·3267
48·6	2361·96	114791·256	6·9714	3·6493	82	6724	551368	9·0554	4·3445
48·7	2371·69	115501·303	6·9785	3·6518	83	6889	571787	9·1104	4·3621
48·8	2381·44	116214·272	6·9857	3·6543	84	7056	592704	9·1652	4·3795
48·9	2391·21	116930·169	6·9929	3·6568	85	7225	614125	9·2195	4·3968
49	2401	117649	7	3·6593	86	7396	636056	9·2736	4·414
49·1	2410·81	118370·771	7·0071	3·6618	87	7569	658503	9·3274	4·431
49·2	2420·64	119095·488	7·0143	3·6643	88	7744	681472	9·3808	4·448
49·3	2430·49	119823·157	7·0214	3·6668	89	7921	704969	9·434	4·4647
49·4	2440·36	120553·784	7·0285	3·6692	90	8100	729000	9·4868	4·4814
49·5	2450·25	121287·375	7·0356	3·6717	91	8281	753571	9·5394	4·4979
49·6	2460·16	122023·936	7·0427	3·6742	92	8464	778688	9·5917	4·5144
49·7	2470·09	122763·473	7·0498	3·6766	93	8649	804357	9·6437	4·5307
49·8	2480·04	123505·992	7·0569	3·6791	94	8836	830584	9·6954	4·5468
49·9	2490·01	124251·499	7·064	3·6816	95	9025	857375	9·7468	4·5629
50	2500	125000	7·0711	3·684	96	9216	884736	9·798	4·5789
51	2601	132651	7·1414	3·7084	97	9409	912673	9·8489	4·5947
52	2704	140608	7·2111	3·7325	98	9604	941192	9·8995	4·6104
53	2809	148877	7·2801	3·7563	99	9801	970299	9·9499	4·6261

TABLE 26.

Square, Cube, and Fourth Roots (Fractional).

(Values intermediate between those given may, if desired, be interpolated by simple proportion.)

No.	$\sqrt{}$ Sq. root.	$\sqrt[3]{}$ Cube root.	$\sqrt[4]{}$ 4th root.	No.	$\sqrt{}$ Sq. root.	$\sqrt[3]{}$ Cube root.	$\sqrt[4]{}$ 4th root.
0·1	0·316	0·464	0·5622	0·45	0·671	0·766	0·8192
0·11	0·3317	0·4791	0·5759	0·46	0·6782	0·7719	0·82355
0·12	0·3464	0·4932	0·5885	0·47	0·6856	0·7775	0·828
0·13	0·3606	0·5066	0·6004	0·48	0·6928	0·783	0·8323
0·14	0·3741	0·5192	0·6117	0·49	0·7	0·7884	0·8367
0·15	0·387	0·531	0·6227	0·50	0·707	0·794	0·8409
0·16	0·4	0·5429	0·6325	0·525	0·7246	0·8067	0·8512
0·17	0·4123	0·554	0·6421	0·550	0·742	0·819	0·8614
0·18	0·4243	0·5646	0·6514	0·575	0·7583	0·8316	0·8708
0·19	0·4359	0·5749	0·6602	0·6	0·775	0·843	0·8804
0·20	0·447	0·585	0·6686	0·625	0·7906	0·855	0·8891
0·21	0·4583	0·5944	0·677	0·650	0·806	0·866	0·8978
0·22	0·469	0·6037	0·6849	0·675	0·8216	0·8772	0·9064
0·23	0·4796	0·6127	0·6925	0·7	0·837	0·888	0·9149
0·24	0·4899	0·6215	0·6999	0·725	0·8515	0·89835	0·9228
0·25	0·5	0·63	0·7071	0·750	0·866	0·909	0·9306
0·26	0·5099	0·6383	0·7141	0·775	0·8803	0·9185	0·9383
0·27	0·5196	0·6463	0·7208	0·8	0·894	0·928	0·9455
0·28	0·5292	0·6542	0·7274	0·825	0·9083	0·9379	0·9531
0·29	0·5386	0·6619	0·7338	0·850	0·922	0·947	0·9602
0·30	0·548	0·669	0·7403	0·875	0·9354	0·9565	0·9672
0·31	0·5568	0·6768	0·7462	0·9	0·949	0·965	0·9742
0·32	0·5657	0·684	0·7532	0·925	0·9618	0·9743	0·9807
0·33	0·5745	0·691	0·7579	0·950	0·975	0·983	0·9874
0·34	0·5831	0·698	0·7636	0·975	0·9874	0·9916	0·9937
0·35	0·592	0·705	0·7694	1·0	1·000	1·000	1·000
0·36	0·6	0·7114	0·7746	1·05	1·025	1·016	1·0124
0·37	0·6083	0·7179	0·7799	1·1	1·049	1·032	1·0242
0·38	0·6164	0·7243	0·7851	1·15	1·072	1·048	1·0351
0·39	0·6245	0·7306	0·7903	1·2	1·095	1·063	1·0464
0·40	0·633	0·737	0·7957	1·25	1·118	1·077	1·0574
0·41	0·6403	0·7429	0·8002	1·3	1·140	1·091	1·0677
0·42	0·6481	0·7489	0·805	1·35	1·162	1·105	1·078
0·43	0·6557	0·7548	0·8098	1·4	1·183	1·119	1·0876
0·44	0·6633	0·7606	0·81445	1·45	1·204	1·132	1·0973

Table of Roots—(*continued*).

No.	√ Sq. root.	∛ Cube root.	∜ 4th root.	No.	√ Sq. root.	∛ Cube root.	∜ 4th root.
1·5	1·225	1·145	1·1068	5·0	2·236	1·710	1·4953
1·55	1·245	1·157	1·1158	5·1	2·258	1·721	1·5017
1·6	1·265	1·170	1·1247	5·2	2·280	1·733	1·51
1·65	1·285	1·182	1·1336	5·3	2·302	1·744	1·5172
1·7	1·304	1·194	1·142	5·4	2·324	1·754	1·5245
1·75	1·323	1·205	1·1502	5·5	2·345	1·765	1·5313
1·8	1·342	1·216	1·1582	5·6	2·366	1·776	1·5382
1·85	1·360	1·228	1·1662	5·7	2·388	1·786	1·5453
1·9	1·378	1·239	1·174	5·8	2·408	1·797	1·5518
1·95	1·396	1·249	1·1815	5·9	2·429	1·807	1·5585
2·0	1·414	1·260	1·1891	6·0	2·450	1·817	1·5625
2·1	1·449	1·281	1·2038	6·1	2·470	1·827	1·5716
2·2	1·483	1·301	1·2178	6·2	2·490	1·837	1·578
2·3	1·517	1·320	1·2317	6·3	2·510	1·847	1·5843
2·4	1·549	1·339	1·2446	6·4	2·530	1·857	1·5906
2·5	1·581	1·357	1·2574	6·5	2·550	1·866	1·5969
2·6	1·613	1·375	1·27	6·6	2·569	1·876	1·6028
2·7	1·643	1·393	1·2818	6·7	2·588	1·885	1·6089
2·8	1·673	1·409	1·2934	6·8	2·608	1·895	1·615
2·9	1·703	1·426	1·305	6·9	2·627	1·904	1·621
3·0	1·732	1·442	1·316	7·0	2·646	1·913	1·6263
3·1	1·761	1·458	1·327	7·1	2·665	1·922	1·6325
3·2	1·789	1·474	1·338	7·2	2·683	1·931	1·638
3·3	1·817	1·489	1·348	7·3	2·702	1·940	1·6438
3·4	1·844	1·504	1·358	7·4	2·720	1·949	1·6492
3·5	1·871	1·518	1·3678	7·5	2·739	1·957	1·655
3·6	1·897	1·533	1·3773	7·6	2·757	1·966	1·6604
3·7	1·924	1·547	1·387	7·7	2·775	1·975	1·666
3·8	1·949	1·561	1·396	7·8	2·793	1·983	1·6713
3·9	1·975	1·574	1·4053	7·9	2·811	1·992	1·6766
4·0	2·000	1·587	1·4142	8·0	2·828	2·000	1·6817
4·1	2·025	1·601	1·423	8·1	2·846	2·008	1·687
4·2	2·049	1·613	1·4314	8·2	2·864	2·017	1·6923
4·3	2·074	1·626	1·4401	8·3	2·881	2·025	1·6973
4·4	2·098	1·639	1·4484	8·4	2·898	2·033	1·7024
4·5	2·121	1·651	1·4563	8·5	2·916	2·041	1·7076
4·6	2·145	1·663	1·4646	8·6	2·933	2·049	1·7126
4·7	2·168	1·675	1·4724	8·7	2·950	2·057	1·7176
4·8	2·191	1·687	1·4802	8·8	2·967	2·065	1·7225
4·9	2·214	1·699	1·488	8·9	2·983	2·072	1·7271

TABLE OF ROOTS—(*continued*).

No.	$\sqrt{}$ Sq. root.	$\sqrt[3]{}$ Cube root.	$\sqrt[4]{}$ 4th root.	No.	$\sqrt{}$ Sq. root.	$\sqrt[3]{}$ Cube root.	$\sqrt[4]{}$ 4th root.
9·0	3·000	2·080	1·732	12·9	3·592	2·345	1·8952
9·1	3·017	2·088	1·737	13·0	3·606	2·351	1·8990
9·2	3·033	2·095	1·7415	13·2	3·633	2·363	1·906
9·3	3·050	2·103	1·7464	13·4	3·661	2·375	1·9134
9·4	3·066	2·111	1·751	13·6	3·668	2·387	1·9204
9·5	3·082	2·118	1·7555	13·8	3·715	2·399	1·9274
9·6	3·098	2·125	1·7601	14·0	3·742	2·410	1·9344
9·7	3·115	2·133	1·765	14·2	3·768	2·422	1·9411
9·8	3·131	2·14	1·7695	14·4	3·795	2·433	1·9481
9·9	3·146	2·147	1·7737	14·6	3·821	2·444	1·9547
10·0	3·162	2·154	1·7782	14·8	3·847	2·455	1·9614
10·1	3·178	2·162	1·7827	15·0	3·873	2·466	1·968
10·2	3·194	2·169	1·7872	15·2	3·899	2·477	1·9746
10·3	3·209	2·177	1·7914	15·4	3·924	2·488	1·9809
10·4	3·225	2·183	1·7958	15·6	3·950	2·499	1·9875
10·5	3·240	2·189	1·8000	15·8	3·975	2·509	1·9937
10·6	3·256	2·197	1·8044	16·0	4·000	2·520	2·0000
10·7	3·271	2·204	1·8086	16·2	4·025	2·530	2·0062
10·8	3·286	2·211	1·8127	16·4	4·050	2·541	2·0125
10·9	3·302	2·217	1·8171	16·6	4·074	2·551	2·0184
11·0	3·317	2·224	1·8213	16·8	4·099	2·561	2·0246
11·1	3·332	2·231	1·8253	17·0	4·123	2·571	2·0305
11·2	3·347	2·237	1·8295	17·2	4·147	2·581	2·0364
11·3	3·362	2·244	1·8336	17·4	4·171	2·591	2·0423
11·4	3·376	2·251	1·8374	17·6	4·195	2·601	2·0481
11·5	3·391	2·257	1·8415	17·8	4·219	2·611	2·054
11·6	3·406	2·264	1·8455	18·0	4·243	2·621	2·0599
11·7	3·421	2·270	1·8496	18·2	4·266	2·630	2·0654
11·8	3·435	2·277	1·8534	18·4	4·290	2·640	2·0712
11·9	3·450	2·283	1·8574	18·6	4·313	2·650	2·0768
12·0	3·464	2·289	1·8612	18·8	4·336	2·659	2·0823
12·1	3·479	2·296	1·8652	19·0	4·359	2·668	2·0878
12·2	3·493	2·302	1·8689	19·2	4·382	2·678	2·0933
12·3	3·507	2·308	1·8727	19·4	4·405	2·687	2·0988
12·4	3·521	2·315	1·8764	19·6	4·427	2·696	2·104
12·5	3·536	2·321	1·8804	19·8	4·450	2·705	2·1095
12·6	3·550	2·327	1·8841	20·0	4·472	2·714	2·1147
12·7	3·564	2·333	1·8879	20·2	4·494	2·723	2·12
12·8	3·578	2·339	1·8915	20·4	4·517	2·732	2·125

TABLE 27.

Whitworth's Standard 55° Screw Threads for Bolts.

(With sizes of hexagonal nuts and bolt heads.)

Diameter of bolt. Fractional sizes.	Diameter of bolt. Decimal sizes.	Number of threads per inch.	Diameter at bottom of thread.	Distance across flats.	Distance across corners.	Thickness of bolt head.	Thickness of nut.
1/16	·0625	60	·0411	·212	·2447	·0547	1/16
3/32	·09375	48	·0670	·280	·3233	·0820	3/32
1/8	·125	40	·0929	·338	·3902	·1093	1/8
3/16	·1875	24	·1341	·448	·5173	·1640	3/16
1/4	·25	20	·1859	·525	·6062	·2187	1/4
5/16	·3125	18	·2413	·6014	·6944	·2734	5/16
3/8	·375	16	·2949	·7094	·8191	·3281	3/8
7/16	·4375	14	·3460	·8204	·9473	·3828	7/16
1/2	·5	12	·3932	·9191	1·0612	·4375	1/2
9/16	·5625	12	·4557	1·011	1·1674	·4921	9/16
5/8	·625	11	·5085	1·101	1·2713	·5468	5/8
11/16	·6875	11	·5710	1·2011	1·3869	·6015	11/16
3/4	·75	10	·6219	1·3012	1·5024	·6562	3/4
13/16	·8125	10	·6844	1·39	1·6050	·7109	13/16
7/8	·875	9	·7327	1·4788	1·7075	·7656	7/8
15/16	·9375	9	·7952	1·5745	1·8180	·8203	15/16
1	1·0	8	·8399	1·6701	1·9284	·875	1
1 1/8	1·125	7	·9420	1·8605	2·1483	·9843	1 1/8
1 1/4	1·25	7	1·0670	2·0483	2·3651	1·0937	1 1/4
1 3/8	1·375	6	1·1615	2·2146	2·5571	1·2031	1 3/8
1 1/2	1·5	6	1·2865	2·4134	2·7867	1·3125	1 1/2
1 5/8	1·625	5	1·3688	2·5763	2·9748	1·4218	1 5/8
1 3/4	1·75	5	1·4938	2·7578	3·1844	1·5312	1 3/4
1 7/8	1·875	4·5	1·5904	3·0183	3·4852	1·6406	1 7/8
2	2·0	4·5	1·7154	3·1491	3·6362	1·75	2
2 1/8	2·125	4·5	1·8404	3·337	3·8532	1·8593	2 1/8
2 1/4	2·25	4	1·9298	3·546	4·0945	1·9687	2 1/4
2 3/8	2·375	4	2·0548	3·75	4·3301	2·0781	2 3/8
2 1/2	2·5	4	2·1798	3·894	4·4964	2·1875	2 1/2
2 5/8	2·625	4	2·3048	4·049	4·6753	2 2968	2 5/8
2 3/4	2·75	3·5	2·3840	4·181	4·8278	2·4062	2 3/4
2 7/8	2·875	3·5	2·5090	4·3456	5·0178	2·5156	2 7/8
3	3·0	3·5	2·6340	4·531	5·2319	2·625	3
3 1/8	3·125	3·5	2·7590	4·69	5·4155	2·734	3 1/8
3 1/4	3·25	3·25	2·8559	4·85	5·6002	2·843	3 1/4
3 3/8	3·375	3·25	2·9809	5·01	5·7850	2·953	3 3/8
3 1/2	3·5	3·25	3·1059	5·175	5·9755	3·062	3 1/2
3 5/8	3·625	3·25	3·2309	5·362	6·1915	3·171	3 5/8
3 3/4	3·75	3	3·3231	5·55	6·4085	3·281	3 3/4
3 7/8	3·875	3	3·4481	5·75	6·6395	3·39	3 7/8

WHITWORTH'S STANDARD SCREW THREADS, ETC.—(*continued*).

Diameter of bolt.		Number of threads per inch.	Diameter at bottom of thread.	Distance across flats.	Distance across corners.	Thickness of bolt head.	Thickness of nut.
Fractional sizes.	Decimal sizes.						
4	4·0	3	3·5731	5·95	6·8704	3·5	4
4⅛	4·125	3	3·6981	6·162	7·1152	3·609	4⅛
4¼	4·25	2·875	3·8045	6·375	7·3612	3·718	4¼
4⅜	4·375	2·875	3·9295	6·6	7·6210	3·828	4⅜
4½	4·5	2·875	4·0545	6·825	7·8819	3·937	4½
4⅝	4·625	2·875	4·1795	7·0625	8·1550	4·046	4⅝
4¾	4·75	2·75	4·2843	7·3	8·4293	4·156	4¾
4⅞	4·875	2·75	4·4093	7·55	8·7179	4·265	4⅞
5	5·0	2·75	4·5343	7·8	9·0066	4·375	5
5⅛	5·125	2·75	4·6593	8·065	9·3126	4·484	5⅛
5¼	5·25	2·625	4·7621	8·35	9·6417	4·593	5¼
5⅜	5·375	2·625	4·8871	8·6	9·9304	4·703	5⅜
5½	5·5	2·625	5·0121	8·85	10·2190	4·812	5½
5⅝	5·625	2·625	5·1371	9·15	10·5655	4·921	5⅝
5¾	5·75	2·5	5·2377	9·45	10·9119	5·031	5¾
5⅞	5·875	2·5	5·3627	9·75	10·2583	5·140	5⅞
6	6·0	2·5	5·4877	10	11·5470	5·25	6

For gas and water pipes the Whitworth thread is unsuitable, and for this purpose the standard thread is much shallower, with a much finer pitch. The table below is the standard used in this country for that purpose:—

TABLE 28.

WHITWORTH THREADS FOR GAS AND WATER PIPES.

Internal diameter of pipe.	Diameter at top of thread.	Diameter at bottom of thread.	Number of threads per inch.	Internal diameter of pipe.	Diameter at top of thread.	Diameter at bottom of thread.	Number of threads per inch.
⅛	·3825	·3367	28	1⅞	2·245	2·1285	11
¼	·5180	·4506	19	2	2·347	2·2305	11
⅜	·6563	·5889	19	2⅛	2·467	2·3505	11
½	·8257	·7342	14	2¼	2·5875	2·4710	11
⅝	·9022	·8107	14	2⅜	2·794	2·6775	11
¾	1·041	·9495	14	2½	3·0013	2·8848	11
⅞	1·189	1·0975	14	2⅝	3·124	3·0075	11
1	1·309	1·1925	11	2¾	3·247	3·1305	11
1⅛	1·492	1·3755	11	2⅞	3·367	3·2505	11
1¼	1·650	1·5335	11	3	3·485	3·3685	11
1⅜	1·745	1·6285	11	3¼	3·6985	3·5820	11
1½	1·8825	1·7660	11	3½	3·912	3·7955	11
1⅝	2·021	1·9045	11	3¾	4·1255	4·0090	11
1¾	2·047	1·9305	11	4	4·339	4·2225	11

TABLE 29.

Whitworth Screw Threads for Hydraulic Iron Piping.

The internal and external diameters being those adopted by Messrs. James Russell and Sons.

Diameter of piping. Internal.	Diameter of piping. External.	Diameter at bottom of thread.	Pressure in lbs. per square inch.	Number of threads per inch.
$\frac{1}{4}$	$\frac{5}{8}$	·5335	4000	14
	$\frac{3}{4}$	·6585	6000	
	$\frac{7}{8}$	·7835	8000	
	1	·9085	10000	
$\frac{3}{8}$	$\frac{3}{4}$	·6585	4000	14
	$\frac{7}{8}$	·7835	6000	
	1	·9085	8000	
	$1\frac{1}{8}$	1·0335	10000	
$\frac{1}{2}$	1	·9085	4000	14
	$1\frac{1}{8}$	1·0335	6000	
	$1\frac{1}{4}$	1·1335	8000	11
	$1\frac{3}{8}$	1·2585	10000	
$\frac{5}{8}$	$1\frac{1}{8}$	1·0335	4000	14
	$1\frac{1}{4}$	1·1335	6000	11
	$1\frac{3}{8}$	1·2585	8000	
	$1\frac{1}{2}$	1·3835	10000	
$\frac{3}{4}$	$1\frac{1}{4}$	1·1335	4000	11
	$1\frac{3}{8}$	1·2585	6000	
	$1\frac{1}{2}$	1·3835	8000	
	$1\frac{5}{8}$	1·5085	10000	
$\frac{7}{8}$	$1\frac{3}{8}$	1·2585	4000	11
	$1\frac{1}{2}$	1·3835	6000	
	$1\frac{5}{8}$	1·5085	8000	
	$1\frac{3}{4}$	1·6335	10000	
1	$1\frac{1}{2}$	1·3835	4000	11
	$1\frac{5}{8}$	1·5085	6000	
	$1\frac{3}{4}$	1·6335	8000	
	$1\frac{7}{8}$	1·7585	10000	
$1\frac{1}{8}$	$1\frac{5}{8}$	1·5085	4000	11
	$1\frac{3}{4}$	1·6335	6000	
	$1\frac{7}{8}$	1·7585	8000	
	2	1·8835	10000	

Diameter of piping. Internal.	Diameter of piping. External.	Diameter at bottom of thread.	Pressure in lbs. per square inch.	Number of threads per inch.
$1\frac{1}{4}$	$1\frac{3}{4}$	1·6335	4000	11
	$1\frac{7}{8}$	1·7585	6000	
	2	1·8835	8000	
	$2\frac{1}{8}$	2·0085	10000	
$1\frac{3}{8}$	$1\frac{7}{8}$	1·7585	4000	11
	2	1·8835	6000	
	$2\frac{1}{8}$	2·0085	8000	
	$2\frac{1}{4}$	2·1335	10000	
$1\frac{1}{2}$	2	1·8835	4000	11
	$2\frac{1}{8}$	2·0085	6000	
	$2\frac{1}{4}$	2·1335	8000	
	$2\frac{3}{8}$	2·2585	10000	
	$2\frac{1}{2}$	2·3835	10000	
$1\frac{5}{8}$	$2\frac{1}{8}$	2·0085	4000	11
	$2\frac{1}{4}$	2·1335	6000	
	$2\frac{3}{8}$	2·2585	8000	
	$2\frac{1}{2}$	2·3835	10000	
$1\frac{3}{4}$	$2\frac{1}{4}$	2·1335	3000	11
	$2\frac{3}{8}$	2·2585	4000	
	$2\frac{1}{2}$	2·3835	6000	
	$2\frac{5}{8}$	2·5085	8000	
	$2\frac{3}{4}$	2·6335	10000	
$1\frac{7}{8}$	$2\frac{3}{8}$	2·2585	3000	11
	$2\frac{1}{2}$	2·3835	4000	
	$2\frac{5}{8}$	2·5085	6000	
	$2\frac{3}{4}$	2·6335	8000	
	$2\frac{7}{8}$	2·7585	10000	
2	$2\frac{1}{2}$	2·3835	3000	11
	$2\frac{5}{8}$	2·5085	4000	
	$2\frac{3}{4}$	2·6335	6000	
	$2\frac{7}{8}$	2·7585	8000	
	3	2·8835	10000	

TABLE 30.

United States Standard 60° Screw Threads.

Diameter of screw.	Threads per inch.	Diameter at root of thread.	Width of flat at top and bottom of thread.
1/4	20	·185	·0062
5/16	18	·2403	·0069
3/8	16	·2936	·0078
7/16	14	·3447	·0089
1/2	13	·4001	·0096
9/16	12	·4542	·0104
5/8	11	·5069	·0114
3/4	10	·6201	·0125
7/8	9	·7307	·0139
1	8	·8376	·0156
1 1/8	7	·9394	·0179
1 1/4	7	1·0644	·0179
1 3/8	6	1·1585	·0208
1 1/2	6	1·2835	·0208
1 5/8	5 1/2	1·3888	·0227
1 3/4	5	1·4902	·0250
1 7/8	5	1·6152	·0250
2	4 1/2	1·7113	·0278
2 1/4	4 1/2	1·9613	·0278
2 1/2	4	2·1752	·0313
2 3/4	4	2·4252	·0313
3	3 1/2	2·6288	·0357
3 1/4	3 1/2	2·8788	·0357
3 1/2	3 1/4	3·1003	·0385
3 3/4	3	3·3170	·0417
4	3	3·5670	·0417
4 1/4	2 7/8	3·7982	·0435
4 1/2	2 3/4	4·0276	·0455
4 3/4	2 5/8	4·2551	·0476
5	2 1/2	4·4804	·0500
5 1/4	2 1/2	4·7304	·0500
5 1/2	2 3/8	4·9530	·0526
5 3/4	2 3/8	5·2030	·0526
6	2 1/4	5·4226	·0556

TABLE 31.

International Standard 60° Thread.
(Metric system.)

Diameter of the screw.	Pitch.	Diameter at root of thread.	Width of flat at top and bottom of thread.
Millimetres.	Millimetres.	Millimetres.	Millimetres.
3	·5	2·35	·06
4	·75	3·03	·09
5	·75	4·03	·09
6	1·0	4·70	·13
7	1·0	5·70	·13
8	1·0	6·70	·13
8	1·25	6·38	·16
9	1·0	7·70	·13
9	1·25	7·38	·16
10	1·5	8·05	·19
11	1·5	9·05	·19
12	1·5	10·05	·19
12	1·75	9·73	·22
14	2·0	11·40	·25
16	2·0	13·40	·25
18	2·5	14·75	·31
20	2·5	16·75	·31
22	2·5	18·75	·31
22	3·0	18·10	·38
24	3·0	20·10	·38
26	3·0	22·10	·38
27	3·0	23·10	·38
28	3·0	24·10	·38
30	3·5	25·45	·44
32	3·5	27·45	·44
33	3·5	28·45	·44
34	3·5	29·45	·44
36	4·0	30·80	·5
38	4·0	32·80	·5
39	4·0	33·80	·5
40	4·0	34·80	·5
42	4·5	36·15	·56
44	4·5	38·15	·56
45	4·5	39·15	·56
46	4·5	40·15	·56
48	5·0	41·51	·63
50	5·0	43·51	·63
52	5·0	45·51	·63
56	5·5	48·86	·69
60	5·5	52·86	·69
64	6·0	56·21	·75
68	6·0	60·21	·75
72	6·5	63·56	·81
76	6·5	67·56	·81
80	7·0	70·91	·88

TABLE 32.

British Association 47½° Standard Thread.

This is adopted as the standard screw gauge by the Post-office Telegraphs Department and most large electrical firms.

No.	Nominal dimensions in thousandths of an inch.		Threads per inch.	Absolute dimensions in millimetres.	
	Diameter.	Pitch.		Diameter.	Pitch.
25	10	2·8	353	·25	·072
24	11	3·1	317	·29	·080
23	13	3·5	285	·33	·089
22	15	3·9	259	·37	·098
21	17	4·3	231	·42	·11
20	19	4·7	212	·48	·12
19	21	5·5	181	·54	·14
18	24	5·9	169	·62	·15
17	27	6·7	149	·70	·17
16	31	7·5	134	·79	·19
15	35	8·3	121	·90	·21
14	39	9·1	110	1·0	·23
13	44	9·8	101	1·2	·25
12	51	11·0	90·7	1·3	·28
11	59	12·2	81·9	1·5	·31
10	67	13·8	72·6	1·7	·35
9	75	15·4	65·1	1·9	·39
8	86	16·9	59·1	2·2	·43
7	98	18·9	52·9	2·5	·48
6	100	20·9	47·9	2·8	·53
5	126	23·2	43·0	3·2	·59
4	142	26·0	38·5	3·6	·66
3	161	28·7	34·8	4·1	·73
2	185	31·9	31·4	4·7	·81
1	209	35·4	28·2	5·3	·90
0	236	39·4	25·4	6·0	1·00

TABLE 33.

French Standard 60° Thread.

Diameter.	Pitch.	Diameter.	Pitch.	Diameter.	Pitch.
Millimetres.	Millimetres.	Millimetres.	Millimetres.	Millimetres.	Millimetres.
3	·5	16	2·0	36	4·0
4	·75	18	2·5	38	4·0
5	·75	20	2·5	40	4·0
6	1·0	22	3·0	42	4·5
7	1·0	24	3·0	44	4·5
8	1·0	26	3·0	46	4·5
9	1·0	28	3·0	48	5·0
10	1·5	30	3·5	50	5·0
12	1·5	32	3·5		
14	2·0	34	3·5		

TABLE 34.

Sharp "V" Thread 60°.

Diameter.	Number of threads per inch.	Diameter.	Number of threads per inch.	Diameter.	Number of threads per inch.	Diameter.	Number of threads per inch.
$\frac{1}{4}$	20	$\frac{7}{8}$	9	2	$4\frac{1}{2}$	$3\frac{1}{4}$	$3\frac{1}{2}$
$\frac{5}{16}$	18	$\frac{15}{16}$	9	$2\frac{1}{8}$	$4\frac{1}{2}$	$3\frac{3}{8}$	$3\frac{1}{4}$
$\frac{3}{8}$	16	1	8	$2\frac{1}{4}$	$4\frac{1}{2}$	$3\frac{1}{2}$	$3\frac{1}{4}$
$\frac{7}{16}$	14	$1\frac{1}{8}$	7	$2\frac{3}{8}$	$4\frac{1}{2}$	$3\frac{5}{8}$	$3\frac{1}{4}$
$\frac{1}{2}$	12	$1\frac{1}{4}$	7	$2\frac{1}{2}$	4	$3\frac{3}{4}$	3
$\frac{9}{16}$	12	$1\frac{3}{8}$	6	$2\frac{5}{8}$	4	$3\frac{7}{8}$	3
$\frac{5}{8}$	11	$1\frac{1}{2}$	6	$2\frac{3}{4}$	4	4	3
$\frac{11}{16}$	11	$1\frac{5}{8}$	5	$2\frac{7}{8}$	4		
$\frac{3}{4}$	10	$1\frac{3}{4}$	5	3	$3\frac{1}{2}$		
$\frac{13}{16}$	10	$1\frac{7}{8}$	$4\frac{1}{2}$	$3\frac{1}{8}$	$3\frac{1}{2}$		

TABLE 35.

Table of Decimal Equivalents

Of eighths, sixteenths, thirty-seconds, and sixty-fourths of an inch.

Eighths.		Thirty-seconds.		Sixty-fourths.		Sixty-fourths.	
1/8	·125	1/32	·03125	1/64	·015625	33/64	·515625
1/4	·250	3/32	·09375	3/64	·046875	35/64	·546875
3/8	·375	5/32	·15675	5/64	·078125	37/64	·578125
1/2	·500	7/32	·21875	7/64	·109375	39/64	·609375
5/8	·625	9/32	·28125	9/64	·140625	41/64	·640625
3/4	·750	11/32	·34375	11/64	·171875	43/64	·671875
7/8	·875	13/32	·40625	13/64	·203125	45/64	·703125
Sixteenths.		15/32	·46875	15/64	·234375	47/64	·734375
		17/32	·53125	17/64	·265625	49/64	·765625
1/16	·0625	19/32	·59375	19/64	·296875	51/64	·796875
3/16	·1875	21/32	·65625	21/64	·328125	53/64	·828125
5/16	·3125	23/32	·71875	23/64	·359375	55/64	·859375
7/16	·4375	25/32	·78125	25/64	·390625	57/64	·890625
9/16	·5625	27/32	·84375	27/64	·421875	59/64	·921875
11/16	·6875	29/32	·90625	29/64	·453125	61/64	·953125
13/16	·8125	31/32	·96875	31/64	·484375	63/64	·984375
15/16	·9375						

TABLE 36.

Inches and Fractions of Inches to Millimetres.

Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.
1/32	·79	23/32	18·25	1 13/32	35·71	2 3/32	53·18
1/16	1·58	3/4	19·04	1 7/16	36·51	2 1/8	53·97
3/32	2·38	25/32	19·84	1 15/32	37·30	2 5/32	54·76
1/8	3·17	13/16	20·63	1 1/2	38·09	2 3/16	55·56
5/32	3·96	27/32	21·43	1 17/32	38·89	2 7/32	56·35
3/16	4·76	7/8	22·22	1 9/16	39·68	2 1/4	57·14
7/32	5·55	29/32	23·01	1 19/32	40·48	2 9/32	57·94
1/4	6·34	15/16	23·81	1 5/8	41·27	2 5/16	58·73
9/32	7·14	31/32	24·60	1 21/32	42·06	2 11/32	59·53
5/16	7·93	1	25·39	1 11/16	42·86	2 3/8	60·32
11/32	8·73	1 1/32	26·19	1 23/32	43·65	2 13/32	61·11
3/8	9·52	1 1/16	26·98	1 3/4	44·44	2 7/16	61·91
13/32	10·31	1 3/32	27·78	1 25/32	45·24	2 15/32	62·70
7/16	11·11	1 1/8	28·57	1 13/16	46·03	2 1/2	63·49
15/32	11·90	1 5/32	29·36	1 27/32	46·83	2 17/32	64·29
1/2	12·69	1 3/16	30·16	1 7/8	47·62	2 9/16	65·08
17/32	13·49	1 7/32	30·95	1 29/32	48·41	2 19/32	65·88
9/16	14·28	1 1/4	31·74	1 15/16	49·21	2 5/8	66·67
19/32	15·08	1 9/32	32·54	1 31/32	50·00	2 21/32	67·46
5/8	15·87	1 5/16	33·33	2	50·79	2 11/16	68·26
21/32	16·66	1 11/32	34·13	2 1/32	51·59	2 23/32	69·05
11/16	17·46	1 3/8	34·92	2 1/16	52·38	2 3/4	69·84

Inches to Millimetres—(*continued*).

Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.
2 25/32	70·64	3 27/32	97·63	4 29/32	124·61	5 31/32	151·60
2 13/16	71·43	3 7/8	98·42	4 15/16	125·41	6	152·39
2 27/32	72·23	3 29/32	99·21	4 31/32	126·20	6 1/32	153·19
2 7/8	73·02	3 15/16	100·01	5	126·99	6 1/16	153·98
2 29/32	73·81	3 31/32	100·80	5 1/32	127·79	6 3/32	154·78
2 15/16	74·61	4	101·59	5 1/16	128·58	6 1/8	155·57
2 31/32	75·40	4 1/32	102·39	5 3/32	129·38	6 5/32	156·36
3	76·19	4 1/16	103·18	5 1/8	130·17	6 3/16	157·16
3 1/32	76·99	4 3/32	103·98	5 5/32	130·96	6 7/32	157·95
3 1/16	77·78	4 1/8	104·77	5 3/16	131·76	6 1/4	158·74
3 3/32	78·58	4 5/32	105·56	5 7/32	132·55	6 9/32	159·54
3 1/8	79·37	4 3/16	106·36	5 1/4	133·34	6 5/16	160·33
3 5/32	80·16	4 7/32	107·15	5 9/32	134·14	6 11/32	161·13
3 3/16	80·96	4 1/4	107·94	5 5/16	134·93	6 3/8	161·92
3 7/32	81·75	4 9/32	108·74	5 11/32	135·73	6 13/32	162·71
3 1/4	82·54	4 5/16	109·53	5 3/8	136·52	6 7/16	163·51
3 9/32	83·34	4 11/32	110·33	5 13/32	137·31	6 15/32	164·30
3 5/16	84·13	4 3/8	111·12	5 7/16	138·11	6 1/2	165·09
3 11/32	84·93	4 13/32	111·91	5 15/32	138·90	6 17/32	165·89
3 3/8	85·72	4 7/16	112·71	5 1/2	139·69	6 9/16	166·68
3 13/32	86·51	4 15/32	113·50	5 17/32	140·49	6 19/32	167·48
3 7/16	87·31	4 1/2	114·29	5 9/16	141·28	6 5/8	168·27
3 15/32	88·10	4 17/32	115·09	5 19/32	142·08	6 21/32	169·06
3 1/2	88·89	4 9/16	115·88	5 5/8	142·87	6 11/16	169·86
3 17/32	89·69	4 19/32	116·68	5 21/32	143·66	6 23/32	170·65
3 9/16	90·48	4 5/8	117·47	5 11/16	144·46	6 3/4	171·44
3 19/32	91·28	4 21/32	118·26	5 23/32	145·25	6 25/32	172·24
3 5/8	92·07	4 11/16	119·06	5 3/4	146·04	6 13/16	173·03
3 21/32	92·86	4 23/32	119·85	5 25/32	146·84	6 27/32	173·83
3 11/16	93·66	4 3/4	120·64	5 13/16	147·63	6 7/8	174·62
3 23/32	94·45	4 25/32	121·44	5 27/32	148·43	6 29/32	175·41
3 3/4	95·24	4 13/16	122·23	5 7/8	149·22	6 15/16	176·21
3 25/32	96·04	4 27/32	123·03	5 29/32	150·01	6 31/32	177·00
3 13/16	96·83	4 7/8	123·82	5 15/16	150·81	7	177·79

INCHES TO MILLIMETRES—(*continued*).

Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.
$7\frac{1}{32}$	178·59	$8\frac{3}{32}$	205·58	$9\frac{5}{32}$	232·56	$10\frac{7}{32}$	259·55
$7\frac{1}{16}$	179·38	$8\frac{1}{8}$	206·37	$9\frac{3}{16}$	233·36	$10\frac{1}{4}$	260·34
$7\frac{3}{32}$	180·18	$8\frac{5}{32}$	207·16	$9\frac{7}{32}$	234·15	$10\frac{9}{32}$	261·14
$7\frac{1}{8}$	180·97	$8\frac{3}{16}$	207·96	$9\frac{1}{4}$	234·94	$10\frac{5}{16}$	261·93
$7\frac{5}{32}$	181·76	$8\frac{7}{32}$	208·75	$9\frac{9}{32}$	235·74	$10\frac{11}{32}$	262·73
$7\frac{3}{16}$	182·56	$8\frac{1}{4}$	209·54	$9\frac{5}{16}$	236·53	$10\frac{3}{8}$	263·52
$7\frac{7}{32}$	183·35	$8\frac{9}{32}$	210·34	$9\frac{11}{32}$	237·33	$10\frac{13}{32}$	264·31
$7\frac{1}{4}$	184·14	$8\frac{5}{16}$	211·13	$9\frac{3}{8}$	238·12	$10\frac{7}{16}$	265·11
$7\frac{9}{32}$	184·94	$8\frac{11}{32}$	211·93	$9\frac{13}{32}$	238·91	$10\frac{15}{32}$	265·90
$7\frac{5}{16}$	185·73	$8\frac{3}{8}$	212·72	$9\frac{7}{16}$	239·71	$10\frac{1}{2}$	266·69
$7\frac{11}{32}$	186·53	$8\frac{13}{32}$	213·51	$9\frac{15}{32}$	240·50	$10\frac{17}{32}$	267·49
$7\frac{3}{8}$	187·32	$8\frac{7}{16}$	214·31	$9\frac{1}{2}$	241·29	$10\frac{9}{16}$	268·28
$7\frac{13}{32}$	188·11	$8\frac{15}{32}$	215·10	$9\frac{17}{32}$	242·09	$10\frac{19}{32}$	269·08
$7\frac{7}{16}$	188·91	$8\frac{1}{2}$	215·89	$9\frac{9}{16}$	242·88	$10\frac{5}{8}$	269·87
$7\frac{15}{32}$	189·70	$8\frac{17}{32}$	216·69	$9\frac{19}{32}$	243·68	$10\frac{21}{32}$	270·66
$7\frac{1}{2}$	190·49	$8\frac{9}{16}$	217·48	$9\frac{5}{8}$	244·47	$10\frac{11}{16}$	271·46
$7\frac{17}{32}$	191·29	$8\frac{19}{32}$	218·28	$9\frac{21}{32}$	245·26	$10\frac{23}{32}$	272·25
$7\frac{9}{16}$	192·08	$8\frac{5}{8}$	219·07	$9\frac{11}{16}$	246·06	$10\frac{3}{4}$	273·04
$7\frac{19}{32}$	192·88	$8\frac{21}{32}$	219·86	$9\frac{23}{32}$	246·85	$10\frac{25}{32}$	273·84
$7\frac{5}{8}$	193·67	$8\frac{11}{16}$	220·66	$9\frac{3}{4}$	247·64	$10\frac{13}{16}$	274·63
$7\frac{21}{32}$	194·46	$8\frac{23}{32}$	221·45	$9\frac{25}{32}$	248·44	$10\frac{27}{32}$	275·43
$7\frac{11}{16}$	195·26	$8\frac{3}{4}$	222·24	$9\frac{13}{16}$	249·23	$10\frac{7}{8}$	276·22
$7\frac{23}{32}$	196·05	$8\frac{25}{32}$	223·04	$9\frac{27}{32}$	250·03	$10\frac{29}{32}$	277·01
$7\frac{3}{4}$	196·84	$8\frac{13}{16}$	223·83	$9\frac{7}{8}$	250·82	$10\frac{15}{16}$	277·81
$7\frac{25}{32}$	197·64	$8\frac{27}{32}$	224·63	$9\frac{29}{32}$	251·61	$10\frac{31}{32}$	278·60
$7\frac{13}{16}$	198·43	$8\frac{7}{8}$	225·42	$9\frac{15}{16}$	252·41	11	279·39
$7\frac{27}{32}$	199·23	$8\frac{29}{32}$	226·21	$9\frac{31}{32}$	253·20	$11\frac{1}{32}$	280·19
$7\frac{7}{8}$	200·02	$8\frac{15}{16}$	227·01	10	253·99	$11\frac{1}{16}$	280·98
$7\frac{29}{32}$	200·81	$8\frac{31}{32}$	227·80	$10\frac{1}{32}$	254·79	$11\frac{3}{32}$	281·78
$7\frac{15}{16}$	201·61	9	228·59	$10\frac{1}{16}$	255·58	$11\frac{1}{8}$	282·57
$7\frac{31}{32}$	202·40	$9\frac{1}{32}$	229·39	$10\frac{3}{32}$	256·38	$11\frac{5}{32}$	283·36
8	203·19	$9\frac{1}{16}$	230·18	$10\frac{1}{8}$	257·17	$11\frac{3}{16}$	284·16
$8\frac{1}{32}$	203·99	$9\frac{3}{32}$	230·98	$10\frac{5}{32}$	257·96	$11\frac{7}{32}$	284·95
$8\frac{1}{16}$	204·78	$9\frac{1}{8}$	231·77	$10\frac{3}{16}$	258·76	$11\frac{1}{4}$	285·74

INCHES TO MILLIMETRES—(*continued*).

Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.
11 9/32	286·54	12 11/32	313·53	13 13/32	340·51	14 15/32	367·50
11 5/16	287·33	12 3/8	314·32	13 7/16	341·31	14 1/2	368·29
11 11/32	288·13	12 13/32	315·11	13 15/32	342·10	14 17/32	369·09
11 3/8	288·92	12 7/16	315·91	13 1/2	342·89	14 9/16	369·88
11 13/32	289·71	12 15/32	316·70	13 17/32	343·69	14 19/32	370·68
11 7/16	290·51	12 1/2	317·49	13 9/16	344·48	14 5/8	371·47
11 15/32	291·30	12 17/32	318·29	13 19/32	345·28	14 21/32	372·26
11 1/2	292·09	12 9/16	319·08	13 5/8	346·07	14 11/16	373·06
11 17/32	292·89	12 19/32	319·88	13 21/32	346·86	14 23/32	373·85
11 9/16	293·68	12 5/8	320·67	13 11/16	347·66	14 3/4	374·64
11 19/32	294·48	12 21/32	321·46	13 23/32	348·45	14 25/32	375·44
11 5/8	295·27	12 11/16	322·26	13 3/4	349·24	14 13/16	376·23
11 21/32	296·06	12 23/32	323·05	13 25/32	350·04	14 27/32	377·03
11 11/16	296·86	12 3/4	323·84	13 13/16	350·83	14 7/8	377·82
11 23/32	297·65	12 25/32	324·64	13 27/32	351·63	14 29/32	378·61
11 3/4	298·44	12 13/16	325·43	13 7/8	352·42	14 15/16	379·41
11 25/32	299·24	12 27/32	326·23	13 29/32	353·21	14 31/32	380·20
11 13/16	300·03	12 7/8	327·02	13 15/16	354·01	15	380·99
11 27/32	300·83	12 29/32	327·81	13 31/32	354·80	15 1/32	381·79
11 7/8	301·62	12 15/16	328·61	14	355·59	15 1/16	382·58
11 29/32	302·41	12 31/32	329·40	14 1/32	356·39	15 3/32	383·38
11 15/16	303·21	13	330·19	14 1/16	357·18	15 1/8	384·17
11 31/32	304·00	13 1/32	330·99	14 3/32	357·98	15 5/32	384·96
12	304·79	13 1/16	331·78	14 1/8	358·77	15 3/16	385·76
12 1/32	305·59	13 3/32	332·58	14 5/32	359·56	15 7/32	386·55
12 1/16	306·38	13 1/8	333·37	14 3/16	360·36	15 1/4	387·34
12 3/32	307·18	13 5/32	334·16	14 7/32	361·15	15 9/32	388·14
12 1/8	307·97	13 3/16	334·96	14 1/4	361·94	15 5/16	388·93
12 5/32	308·76	13 7/32	335·75	14 9/32	362·74	15 11/32	389·73
12 3/16	309·56	13 1/4	336·54	14 5/16	363·53	15 3/8	390·52
12 7/32	310·35	13 9/32	337·34	14 11/32	364·33	15 13/32	391·31
12 1/4	311·14	13 5/16	338·13	14 3/8	365·12	15 7/16	392·11
12 9/32	311·94	13 11/32	338·93	14 13/32	365·91	15 15/32	392·90
12 5/16	312·73	13 3/8	339·72	14 7/16	366·71	15 1/2	393·69

INCHES TO MILLIMETRES—(*continued*).

Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.
15 17/32	394·49	16 19/32	421·48	17 21/32	448·46	18 23/32	475·45
15 9/16	395·28	16 5/8	422·27	17 11/16	449·26	18 3/4	476·24
15 19/32	396·08	16 21/32	423·06	17 23/32	450·05	18 25/32	477·04
15 5/8	396·87	16 11/16	423·86	17 3/4	450·84	18 13/16	477·83
15 21/32	397·66	16 23/32	424·65	17 25/32	451·64	18 27/32	478·63
15 11/16	398·46	16 3/4	425·44	17 13/16	452·43	18 7/8	479·42
15 23/32	399·25	16 25/32	426·24	17 27/32	453·23	18 29/32	480·21
15 3/4	400·04	16 13/16	427·03	17 7/8	454·02	18 15/16	481·01
15 25/32	400·84	16 27/32	427·83	17 29/32	454·81	18 31/32	481·80
15 13/16	401·63	16 7/8	428·62	17 15/16	455·61	19	482·59
15 27/32	402·43	16 29/32	429·41	17 31/32	456·40	19 1/32	483·39
15 7/8	403·22	16 15/16	430·21	18	457·19	19 1/16	484·18
15 29/32	404·01	16 31/32	431·00	18 1/32	457·99	19 3/32	484·98
15 15/16	404·81	17	431·79	18 1/16	458·78	19 1/8	485·77
15 31/32	405·60	17 1/32	432·59	18 3/32	459·58	19 5/32	486·56
16	406·39	17 1/16	433·38	18 1/8	460·37	19 3/16	487·36
16 1/32	407·19	17 3/32	434·18	18 5/32	461·16	19 7/32	488·15
16 1/16	407·98	17 1/8	434·97	18 3/16	461·96	19 1/4	488·94
16 3/32	408·78	17 5/32	435·76	18 7/32	462·75	19 9/32	489·74
16 1/8	409·57	17 3/16	436·56	18 1/4	463·54	19 5/16	490·53
16 5/32	410·36	17 7/32	437·35	18 9/32	464·34	19 11/32	491·33
16 3/16	411·16	17 1/4	438·14	18 5/16	465·13	19 3/8	492·12
16 7/32	411·95	17 9/32	438·94	18 11/32	465·93	19 13/32	492·91
16 1/4	412·74	17 5/16	439·73	18 3/8	466·72	19 7/16	493·71
16 9/32	413·54	17 11/32	440·53	18 13/32	467·51	19 15/32	494·50
16 5/16	414·33	17 3/8	441·32	18 7/16	468·31	19 1/2	495·29
16 11/32	415·13	17 13/32	442·11	18 15/32	469·10	19 17/32	496·09
16 3/8	415·92	17 7/16	442·91	18 1/2	469·89	19 9/16	496·88
16 13/32	416·71	17 15/32	443·70	18 17/32	470·69	19 19/32	497·68
16 7/16	417·51	17 1/2	444·49	18 9/16	471·48	19 5/8	498·47
16 15/32	418·30	17 17/32	445·29	18 19/32	472·28	19 21/32	499·26
16 1/2	419·09	17 9/16	446·08	18 5/8	473·07	19 11/16	500·06
16 17/32	419·89	17 19/32	446·88	18 21/32	473·86	19 23/32	500·85
16 9/16	420·68	17 5/8	447·67	18 11/16	474·66	19 3/4	501·64

INCHES TO MILLIMETRES—(*continued*).

Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.	Inches.	Millimetres.
$19\frac{25}{32}$	502·44	$20\frac{27}{32}$	529·43	$21\frac{29}{32}$	556·41	$22\frac{31}{32}$	583·40
$19\frac{13}{16}$	503·23	$20\frac{7}{8}$	530·22	$21\frac{15}{16}$	557·21	23	584·19
$19\frac{27}{32}$	504·03	$20\frac{29}{32}$	531·01	$21\frac{31}{32}$	558·00	$23\frac{1}{32}$	584·99
$19\frac{7}{8}$	504·82	$20\frac{15}{16}$	531·81	22	558·79	$23\frac{1}{16}$	585·78
$19\frac{29}{32}$	505·61	$20\frac{31}{32}$	532·60	$22\frac{1}{32}$	559·59	$23\frac{3}{32}$	586·58
$19\frac{15}{16}$	506·41	21	533·39	$22\frac{1}{16}$	560·38	$23\frac{1}{8}$	587·37
$19\frac{31}{32}$	507·20	$21\frac{1}{32}$	534·19	$22\frac{3}{32}$	561·18	$23\frac{5}{32}$	588·16
20	507·99	$21\frac{1}{16}$	534·98	$22\frac{1}{8}$	561·97	$23\frac{3}{16}$	588·96
$20\frac{1}{32}$	508·79	$21\frac{3}{32}$	535·78	$22\frac{5}{32}$	562·76	$23\frac{7}{32}$	589·75
$20\frac{1}{16}$	509·58	$21\frac{1}{8}$	536·57	$22\frac{3}{16}$	563·56	$23\frac{1}{4}$	590·54
$20\frac{3}{32}$	510·38	$21\frac{5}{32}$	537·36	$22\frac{7}{32}$	564·35	$23\frac{9}{32}$	591·34
$20\frac{1}{8}$	511·17	$21\frac{3}{16}$	538·16	$22\frac{1}{4}$	565·14	$23\frac{5}{16}$	592·13
$20\frac{5}{32}$	511·96	$21\frac{7}{32}$	538·95	$22\frac{9}{32}$	565·94	$23\frac{11}{32}$	592·93
$20\frac{3}{16}$	512·76	$21\frac{1}{4}$	539·74	$22\frac{5}{16}$	566·73	$23\frac{3}{8}$	593·72
$20\frac{7}{32}$	513·55	$21\frac{9}{32}$	540·54	$22\frac{11}{32}$	567·53	$23\frac{13}{32}$	594·51
$20\frac{1}{4}$	514·34	$21\frac{5}{16}$	541·33	$22\frac{3}{8}$	568·32	$23\frac{7}{16}$	595·31
$20\frac{9}{32}$	515·14	$21\frac{11}{32}$	542·13	$22\frac{13}{32}$	569·11	$23\frac{15}{32}$	596·10
$20\frac{5}{16}$	515·93	$21\frac{3}{8}$	542·92	$22\frac{7}{16}$	569·91	$23\frac{1}{2}$	596·89
$20\frac{11}{32}$	516·73	$21\frac{13}{32}$	543·71	$22\frac{15}{32}$	570·70	$23\frac{17}{32}$	597·69
$20\frac{3}{8}$	517·52	$21\frac{7}{16}$	544·51	$22\frac{1}{2}$	571·49	$23\frac{9}{16}$	598·48
$20\frac{13}{32}$	518·31	$21\frac{15}{32}$	545·30	$22\frac{17}{32}$	572·29	$23\frac{19}{32}$	599·28
$20\frac{7}{16}$	519·11	$21\frac{1}{2}$	546·09	$22\frac{9}{16}$	573·08	$23\frac{5}{8}$	600·07
$20\frac{15}{32}$	519·90	$21\frac{17}{32}$	546·89	$22\frac{19}{32}$	573·88	$23\frac{21}{32}$	600·86
$20\frac{1}{2}$	520·69	$21\frac{9}{16}$	547·68	$22\frac{5}{8}$	574·67	$23\frac{11}{16}$	601·66
$20\frac{17}{32}$	521·49	$21\frac{19}{32}$	548·48	$22\frac{21}{32}$	575·46	$23\frac{23}{32}$	602·45
$20\frac{9}{16}$	522·28	$21\frac{5}{8}$	549·27	$22\frac{11}{16}$	576·26	$23\frac{3}{4}$	603·24
$20\frac{19}{32}$	523·08	$21\frac{21}{32}$	550·06	$22\frac{23}{32}$	577·05	$23\frac{25}{32}$	604·04
$20\frac{5}{8}$	523·87	$21\frac{11}{16}$	550·86	$22\frac{3}{4}$	577·84	$23\frac{13}{16}$	604·83
$20\frac{21}{32}$	524·66	$21\frac{23}{32}$	551·65	$22\frac{25}{32}$	578·64	$23\frac{27}{32}$	605·63
$20\frac{11}{16}$	525·46	$21\frac{3}{4}$	552·44	$22\frac{13}{16}$	579·43	$23\frac{7}{8}$	606·42
$20\frac{23}{32}$	526·25	$21\frac{25}{32}$	553·24	$22\frac{27}{32}$	580·23	$23\frac{29}{32}$	607·21
$20\frac{3}{4}$	527·04	$21\frac{13}{16}$	554·03	$22\frac{7}{8}$	581·02	$23\frac{15}{16}$	608·01
$20\frac{25}{32}$	527·84	$21\frac{27}{32}$	554·83	$22\frac{29}{32}$	581·81	$23\frac{31}{32}$	608·80
$20\frac{13}{16}$	528·63	$21\frac{7}{8}$	555·62	$22\frac{15}{16}$	582·61	24	609·59

TABLE 37.

Equivalent Values of Millimetres and Inches.

Milli-metres.	Inches.	Milli-metres.	Inches.	Milli-metres.	Inches.	Milli-metres.	Inches.
1	·0394	26	1·0236	51	2·0079	76	2·9922
2	·0787	27	1·0630	52	2·0473	77	3·0315
3	·1181	28	1·1024	53	2·0866	78	3·0709
4	·1575	29	1·1417	54	2·1260	79	3·1103
5	·1968	30	1·1811	55	2·1654	80	3·1496
6	·2362	31	1·2205	56	2·2047	81	3·1890
7	·2756	32	1·2598	57	2·2441	82	3·2284
8	·3150	33	1·2992	58	2·2835	83	3·2677
9	·3543	34	1·3386	59	2·3228	84	3·3071
10	·3937	35	1·3780	60	2·3622	85	3·3465
11	·4331	36	1·4173	61	2·4016	86	3·3859
12	·4724	37	1·4567	62	2·4410	87	3·4252
13	·5118	38	1·4961	63	2·4803	88	3·4646
14	·5512	39	1·5354	64	2·5197	89	3·5040
15	·5906	40	1·5748	65	2·5591	90	3·5433
16	·6299	41	1·6142	66	2·5984	91	3·5827
17	·6693	42	1·6536	67	2·6378	92	3·6221
18	·7087	43	1·6929	68	2·6772	93	3·6614
19	·7480	44	1·7323	69	2·7166	94	3·7008
20	·7874	45	1·7717	70	2·7559	95	3·7402
21	·8268	46	1·8110	71	2·7953	96	3·7796
22	·8661	47	1·8504	72	2·8347	97	3·8189
23	·9055	48	1·8898	73	2·8740	98	3·8583
24	·9449	49	1·9291	74	2·9134	99	3·8977
25	·9843	50	1·9685	75	2·9528	100	3·9370

(100 millimetres = 1 decimetre.)

TABLE 38.

Table of Decimal Equivalents of Millimetres and Fractions of Millimetres.

$\frac{1}{100}$ mm. = ·0003937 inch.

Milli-metres.	Inches.	Milli-metres.	Inches.	Milli-metres.	Inches.
$\frac{1}{50}$	·00079	$\frac{26}{50}$	·02047	2	·07874
$\frac{2}{50}$	·00157	$\frac{27}{50}$	·02126	3	·11811
$\frac{3}{50}$	·00236	$\frac{28}{50}$	·02205	4	·15748
$\frac{4}{50}$	·00315	$\frac{29}{50}$	·02283	5	·19685
$\frac{5}{50}$	·00394	$\frac{30}{50}$	·02362	6	·23622
$\frac{6}{50}$	·00472	$\frac{31}{50}$	·02441	7	·27559
$\frac{7}{50}$	·00551	$\frac{32}{50}$	·02520	8	·31496
$\frac{8}{50}$	·00630	$\frac{33}{50}$	·02598	9	·35433
$\frac{9}{50}$	·00709	$\frac{34}{50}$	·02677	10	·39370
$\frac{10}{50}$	·00787	$\frac{35}{50}$	·02756	11	·43307
$\frac{11}{50}$	·00866	$\frac{36}{50}$	·02835	12	·47244
$\frac{12}{50}$	·00945	$\frac{37}{50}$	·02913	13	·51181
$\frac{13}{50}$	·01024	$\frac{38}{50}$	·02992	14	·55118
$\frac{14}{50}$	·01102	$\frac{39}{50}$	·03071	15	·59055
$\frac{15}{50}$	·01181	$\frac{40}{50}$	·03150	16	·62992
$\frac{16}{50}$	·01260	$\frac{41}{50}$	·03228	17	·66929
$\frac{17}{50}$	·01339	$\frac{42}{50}$	·03307	18	·70866
$\frac{18}{50}$	·01417	$\frac{43}{50}$	·03386	19	·74803
$\frac{19}{50}$	·01496	$\frac{44}{50}$	·03465	20	·78740
$\frac{20}{50}$	·01575	$\frac{45}{50}$	·03543	21	·82677
$\frac{21}{50}$	·01654	$\frac{46}{50}$	·03722	22	·86614
$\frac{22}{50}$	·01732	$\frac{47}{50}$	·03701	23	·90551
$\frac{23}{50}$	·01811	$\frac{48}{50}$	·03780	24	·94488
$\frac{24}{50}$	·01890	$\frac{49}{50}$	·03858	25	·98425
$\frac{25}{50}$	·01969	1	·03937	26	1·02362

10 mm. = 1 centimetre = ·3937 inches.
10 cm. = 1 decimetre = 3·937 „
10 dm. = 1 metre . = 39·37 „
25·4 mm. = 1 English inch.

TABLE 39.

Pounds per Square Inch in Kilogrammes per Square Centimetre.

Pounds per square inch.	Kilogrammes per square centimetre.	Pounds per square inch.	Kilogrammes per square centimetre.	Pounds per square inch.	Kilogrammes per square centimetre.	Pounds per square inch.	Kilogrammes per square centimetre.
1	·0703	41	2·8826	81	5·6949	205	14·4129
2	·1406	42	2·9529	82	5·7652	210	14·7645
3	·2109	43	3·0232	83	5·8355	215	15·1160
4	·2812	44	3·0935	84	5·9058	220	15·4675
5	·3515	45	3·1638	85	5·9761	225	15·8191
6	·4218	46	3·2341	86	6·0464	230	16·1706
7	·4921	47	3·3044	87	6·1167	235	16·5221
8	·5624	48	3·3747	88	6·1870	240	16·8737
9	·6328	49	3·4450	89	6·2573	245	17·2252
10	·7031	50	3·5153	90	6·3276	250	17·5767
11	·7734	51	3·5856	91	6·3979	255	17·9283
12	·8437	52	3·6559	92	6·4682	260	18·2798
13	·9140	53	3·7263	93	6·5385	265	18·6313
14	·9843	54	3·7966	94	6·6088	270	18·9829
15	1·0546	55	3·8669	95	6·6722	275	19·3344
16	1·1249	56	3·9372	96	6·7495	280	19·6860
17	1·1952	57	4·0075	97	6·8198	285	20·0375
18	1·2655	58	4·0778	98	6·8901	290	20·3890
19	1·3358	59	4·1481	99	6·9604	295	20·7406
20	1·4062	60	4·2184	100	7·0307	300	21·0921
21	1·4764	61	4·2887	105	7·3822	310	21·7951
22	1·5467	62	4·3590	110	7·7338	320	22·4981
23	1·6171	63	4·4293	115	8·0853	330	23·2012
24	1·6874	64	4·4996	120	8·4368	340	23·9043
25	1·7577	65	4·5699	125	8·7884	350	24·6073
26	1·8280	66	4·6402	130	9·1399	360	25·3104
27	1·8983	67	4·7106	135	9·4914	370	26·0135
28	1·9686	68	4·7809	140	9·8430	380	26·7166
29	2·0389	69	4·8512	145	10·1945	390	27·4196
30	3·1092	70	4·9215	150	10·5460	400	28·1227
31	2·1795	71	4·9918	155	10·8976	410	28·8258
32	2·2498	72	5·0621	160	11·2491	420	29·5288
33	2·3201	73	5·1324	165	11·6006	430	30·2319
34	2·3905	74	5·2027	170	11·9522	440	30·9350
35	2·4607	75	5·2730	175	12·3037	450	31·6380
36	2·5310	76	5·3433	180	12·6553	460	32·3411
37	2·6013	77	5·4136	185	13·0068	470	33·0442
38	2·6717	78	5·4939	190	13·3583	480	33·7437
39	2·7420	79	5·5542	195	13·7099	490	34·4508
40	2·8123	80	5·6246	200	14·0614	500	35·1533

TABLE 40.

Kilogrammes in Pounds.

Kilos.	Pounds.	Kilos.	Pounds.	Kilos.	Pounds.	Kilos.	Pounds.
1	2·205	26	57·320	51	112·436	76	167·551
2	4·409	27	59·525	52	114·640	77	169·756
3	6·614	28	61·729	53	116·845	78	171·960
4	8·818	29	63·934	54	119·049	79	174·165
5	11·023	30	66·139	55	121·254	80	176·370
6	13·228	31	68·343	56	123·459	81	178·574
7	15·432	32	70·548	57	125·663	82	180·779
8	17·637	33	72·752	58	127·868	83	182·983
9	19·842	34	74·957	59	130·073	84	185·118
10	22·046	35	77·162	60	132·277	85	187·393
11	24·251	36	79·366	61	134·482	86	189·597
12	26·455	37	81·571	62	136·486	87	191·802
13	28·660	38	83·776	63	138·891	88	194·010
14	30·865	39	85·980	64	141·096	89	196·211
15	33·069	40	88·185	65	143·300	90	198·416
16	35·274	41	90·389	66	145·505	91	200·620
17	37·479	42	92·594	67	147·710	92	202·825
18	39·683	43	94·799	68	149·914	93	205·030
19	41·888	44	97·003	69	152·119	94	207·234
20	44·092	45	99·208	70	154·323	95	209·439
21	46·297	46	101·413	71	156·528	96	211·644
22	48·502	47	103·617	72	158·733	97	213·848
23	50·706	48	105·822	73	160·937	98	216·053
24	52·911	49	108·026	74	163·142	99	218·275
25	55·115	50	110·231	75	165·347	100	220·462

TABLE 41.

POUNDS IN KILOGRAMMES.

Pounds.	Kilogrs.	Pounds.	Kilogrs.	Pounds.	Kilogrs.	Pounds.	Kilogrs.
1	·454	26	11·793	51	23·133	76	34·473
2	·907	27	12·247	52	23·587	77	34·927
3	1·361	28	12·701	53	24·040	78	35·380
4	1·814	29	13·154	54	24·494	79	35·834
5	2·268	30	13·608	55	24·948	80	36·287
6	2·722	31	14·061	56	25·401	81	36·741
7	3·175	32	14·515	57	25·855	82	37·195
8	3·629	33	14·969	58	26·308	83	37·648
9	4·082	34	15·422	59	26·762	84	38·102
10	4·536	35	15·876	60	27·252	85	38·555
11	4·989	36	16·329	61	27·669	86	39·009
12	5·443	37	16·783	62	28·123	87	39·463
13	5·897	38	17·236	63	28·576	88	39·916
14	6·350	39	17·690	64	29·030	89	40·370
15	6·804	40	18·144	65	29·483	90	40·823
16	7·257	41	18·597	66	29·937	91	41·277
17	7·711	42	19·051	67	30·391	92	41·731
18	8·165	43	19·504	68	30·844	93	42·184
19	8·618	44	19·958	69	31·298	94	42·638
20	9·072	45	20·412	70	31·751	95	43·091
21	9·525	46	20·865	71	32·205	96	43·545
22	9·979	47	21·319	72	32·659	97	43·998
23	10·433	48	21·772	73	33·112	98	44·452
24	10·886	49	22·226	74	33·566	99	44·906
25	11·340	50	22·680	75	34·019	100	45·359

TABLE 42.

Feet and Inches with Millimetre Equivalents.

Ft. Ins.	Millimetres.	Ft. Ins.	Millimetres.	Ft. Ins.	Millimetres.	Ft. Ins.	Millimetres.
0 1	25·39	3 7	1092·2	7 1	2159·0	10 7	3225·8
0 2	50·79	3 8	1117·6	7 2	2184·4	10 8	3251·2
0 3	76·19	3 9	1143·0	7 3	2209·8	10 9	3276·6
0 4	101·59	3 10	1168·4	7 4	2235·2	10 10	3302·0
0 5	126·99	3 11	1193·8	7 5	2260·6	10 11	3327·4
0 6	152·39	4 0	1219·2	7 6	2286·0	11 0	3352·8
0 7	177·79	4 1	1244·6	7 7	2311·4	11 1	3378·2
0 8	203·19	4 2	1270·0	7 8	2336·8	11 2	3403·6
0 9	228·59	4 3	1295·4	7 9	2362·2	11 3	3429·0
0 10	253·99	4 4	1320·8	7 10	2387·6	11 4	3454·4
0 11	279·39	4 5	1346·2	7 11	2413·0	11 5	3479·8
1 0	304·79	4 6	1371·6	8 0	2438·4	11 6	3505·2
1 1	330·19	4 7	1397·0	8 1	2463·8	11 7	3530·6
1 2	355·59	4 8	1422·4	8 2	2489·2	11 8	3556·0
1 3	380·99	4 9	1447·8	8 3	2514·6	11 9	3581·4
1 4	406·39	4 10	1473·2	8 4	2540·0	11 10	3606·8
1 5	431·79	4 11	1498·6	8 5	2565·4	11 11	3632·2
1 6	457·19	5 0	1524·0	8 6	2590·8	12 0	3657·6
1 7	482·59	5 1	1549·4	8 7	2616·2	12 1	3683·0
1 8	507·99	5 2	1574·8	8 8	2641·6	12 2	3708·4
1 9	533·39	5 3	1600·2	8 9	2667·0	12 3	3733·8
1 10	558·79	5 4	1625·6	8 10	2692·4	12 4	3759·2
1 11	584·19	5 5	1651·0	8 11	2717·8	12 5	3784·6
2 0	609·59	5 6	1676·4	9 0	2743·2	12 6	3810·0
2 1	634·99	5 7	1701·8	9 1	2768·6	12 7	3835·4
2 2	660·39	5 8	1727·2	9 2	2794·0	12 8	3860·8
2 3	685·79	5 9	1752·6	9 3	2819·4	12 9	3886·2
2 4	711·19	5 10	1778·0	9 4	2844·8	12 10	3911·6
2 5	736·59	5 11	1803·4	9 5	2870·2	12 11	3937·0
2 6	761·99	6 0	1828·8	9 6	2895·6	13 0	3962·4
2 7	787·39	6 1	1854·2	9 7	2921·0	13 1	3987·8
2 8	812·79	6 2	1879·6	9 8	2946·4	13 2	4013·2
2 9	838·19	6 3	1905·0	9 9	2971·8	13 3	4038·6
2 10	863·59	6 4	1930·4	9 10	2997·2	13 4	4064·0
2 11	888·99	6 5	1955·8	9 11	3022·6	13 5	4089·4
3 0	914·39	6 6	1981·2	10 0	3048·0	13 6	4114·8
3 1	939·79	6 7	2006·6	10 1	3073·4	13 7	4140·2
3 2	965·19	6 8	2032·0	10 2	3098·8	13 8	4165·6
3 3	990·59	6 9	2057·4	10 3	3124·2	13 9	4191·0
3 4	1016·0	6 10	2082·8	10 4	3149·6	13 10	4216·4
3 5	1041·4	6 11	2108·2	10 5	3175·0	13 11	4241·8
3 6	1066·8	7 0	2133·6	10 6	3200·4	14 0	4267·2

Feet and Inches to Millimetres—(*continued*).

Ft.	Ins.	Milli-metres.	Ft.	Ins.	Milli-metres.	Ft.	Ins.	Milli-metres.	Ft.	Ins.	Milli-metres.
14	1	4292·6	15	7	4749·8	17	1	5207·0	18	7	5664·2
14	2	4318·0	15	8	4775·2	17	2	5232·4	18	8	5689·6
14	3	4343·4	15	9	4800·6	17	3	5257·8	18	9	5715·0
14	4	4368·8	15	10	4826·0	17	4	5283·2	18	10	5740·4
14	5	4394·2	15	11	4851·4	17	5	5308·6	18	11	5765·8
14	6	4419·6	16	0	4876·8	17	6	5334·0	19	0	5791·2
14	7	4445·0	16	1	4902·2	17	7	5359·4	19	1	5816·6
14	8	4470·4	16	2	4927·6	17	8	5384·8	19	2	5842·0
14	9	4495·8	16	3	4953·0	17	9	5410·2	19	3	5867·4
14	10	4521·2	16	4	4978·4	17	10	5435·6	19	4	5892·8
14	11	4546·6	16	5	5003·8	17	11	5461·0	19	5	5918·2
15	0	4572·0	16	6	5029·2	18	0	5486·4	19	6	5943·6
15	1	4597·4	16	7	5054·6	18	1	5511·8	19	7	5969·0
15	2	4622·8	16	8	5080·0	18	2	5537·2	19	8	5994·4
15	3	4648·2	16	9	5105·4	18	3	5562·6	19	9	6019·8
15	4	4673·6	16	10	5130·8	18	4	5588·0	19	10	6045·2
15	5	4699·0	16	11	5156·2	18	5	5613·4	19	11	6070·6
15	6	4724·4	17	0	5181·6	18	6	5638·8	20	0	6096·0

TABLE 43.

Lineal Yards in Metres.

Yards.	Metres.	Yards.	Metres.	Yards.	Metres.	Yards.	Metres.	Yards.	Metres.
1	·914	21	19·202	41	37·490	61	55·778	81	74·066
2	1·829	22	20·117	42	38·404	62	56·692	82	74·980
3	2·743	23	21·031	43	39·319	63	57·607	83	75·894
4	3·658	24	21·945	44	40·233	64	58·521	84	76·809
5	4·572	25	22·860	45	41·147	65	59·435	85	77·723
6	5·486	26	23·774	46	42·062	66	60·350	86	78·637
7	6·401	27	24·688	47	42·976	67	61·264	87	79·552
8	7·315	28	25·603	48	43·891	68	62·178	88	80·466
9	8·229	29	26·517	49	44·805	69	63·093	89	81·381
10	9·144	30	27·432	50	45·719	70	64·007	90	82·295
11	10·058	31	28·346	51	46·634	71	64·922	91	83·209
12	10·973	32	29·260	52	47·548	72	65·836	92	84·124
13	11·887	33	30·175	53	48·463	73	66·750	93	85·038
14	12·801	34	31·089	54	49·377	74	67·665	94	85·953
15	13·716	35	32·004	55	50·291	75	68·579	95	86·867
16	14·630	36	32·918	56	51·206	76	69·494	96	87·781
17	15·545	37	33·832	57	52·120	77	70·408	97	88·696
18	16·459	38	34·747	58	53·035	78	71·322	98	89·610
19	17·373	39	35·661	59	53·949	79	72·237	99	90·525
20	18·288	40	36·576	60	54·863	80	73·151	100	91·439

TABLE 44.

Metres in Lineal Yards.

Metres.	Yards.	Metres.	Yards.	Metres.	Yards.
1	1·094	35	38·277	68	74·366
2	2·188	36	39·370	69	75·460
3	3·281	37	40·464	70	76·553
4	4·374	38	41·558	71	77·647
5	5·468	39	42·651	72	78·741
6	6·562	40	43·745	73	79·834
7	7·655	41	44·838	74	80·928
8	8·749	42	45·932	75	82·021
9	9·843	43	47·026	76	83·115
10	10·936	44	48·119	77	84·209
11	12·030	45	49·213	78	85·302
12	13·123	46	50·306	79	86·396
13	14·217	47	51·400	80	87·490
14	15·311	48	52·494	81	88·583
15	16·404	49	53·587	82	89·677
16	17·498	50	54·681	83	90·770
17	18·591	51	55·775	84	91·864
18	19·685	52	56·868	85	92·958
19	20·779	53	57·962	86	94·051
20	21·872	54	59·055	87	95·145
21	23·966	55	60·149	88	96·239
22	24·060	56	61·243	89	97·332
23	25·153	57	62·336	90	98·426
24	26·247	58	63·430	91	99·519
25	27·340	59	64·524	92	100·613
26	28·434	60	65·617	93	101·707
27	29·528	61	66·711	94	102·800
28	30·621	62	67·804	95	103·894
29	31·715	63	68·898	96	104·987
30	32·809	64	69·992	97	106·081
31	33·902	65	71·085	98	107·175
32	34·996	66	72·179	99	108·268
33	36·089	67	73·272	100	109·362
34	37·183				

TABLE 45.

Decimal Fractions of a Lineal Inch in Millimetres.

Inch.	Milli-metres.	Inch.	Milli-metres.	Inch.	Milli-metres.	Inches.	Milli-metres.
·01	·254	·29	7·366	·57	14·478	·85	21·590
·02	·508	·30	7·620	·58	14·732	·86	21·844
·03	·762	·31	7·874	·59	14·986	·87	22·098
·04	1·016	·32	8·128	·60	15·240	·88	22·352
·05	1·270	·33	8·382	·61	15·494	·89	22·606
·06	1·254	·34	8·636	·62	15·748	·90	22·860
·07	1·778	·35	8·890	·63	16·002	·91	23·114
·08	2·032	·36	9·114	·64	16·256	·92	23·368
·09	2·286	·37	9·398	·65	16·510	·93	23·622
·10	2·540	·38	9·652	·66	16·764	·94	23·876
·11	2·794	·39	9·906	·67	17·018	·95	24·130
·12	3·048	·40	10·160	·68	17·272	·96	24·384
·13	3·302	·41	10·414	·69	17·526	·97	24·638
·14	3·556	·42	10·668	·70	17·780	·98	24·892
·15	3·810	·43	10·922	·71	18·034	·99	25·146
·16	4·064	·44	11·176	·72	18·288	1·00	25·400
·17	4·318	·45	11·430	·73	18·542	2·00	50·799
·18	4·572	·46	11·684	·74	18·796	3·00	76·199
·19	4·826	·47	11·938	·75	19·050	4·00	101·598
·20	5·080	·48	12·192	·76	19·304	5·00	126·998
·21	5·334	·49	12·446	·77	19·558	6·00	152·397
·22	5·588	·50	12·700	·78	19·812	7·00	177·797
·23	5·842	·51	12·954	·79	20·066	8·00	203·196
·24	6·096	·52	13·208	·80	20·320	9·00	228·596
·25	6·350	·53	13·462	·81	20·574	10·00	253·995
·26	6·604	·54	13·716	·82	20·828	11·00	279·395
·27	6·858	·55	13·970	·83	21·082	12·00 =1 foot	304·794
·28	7·112	·56	14·224	·84	21·336		

Metric Conversion Tables.

English to Metrical System.

Pounds per foot . .	×	1·488	= kilos. per metre.
Pounds per yard . .	×	·496	= kilos. per metre.
Tons per foot . . .	×	3333·33	= kilos. per metre.
Tons per yard . . .	×	1111·11	= kilos. per metre.
Pounds per mile . .	×	·2818	= kilos. per metre.
Pounds per square inch	×	·0703	= kilos. per square centimetre.
Pounds per square foot	×	4·883	= kilos. per square metre.
Tons per square foot	×	10·236	= tonnes per square metre.
Tons per square yard	×	1·215	= tonnes per square metre.
Pounds per cubic yard	×	·5933	= kilos. per cubic metre.
Pounds per cubic foot	×	16·020	= kilos. per cubic metre.
Tons per cubic yard .	×	1·329	= tonnes per cubic metre.
Grains per gallon . .	×	·01426	= grammes per litre.
Pounds per gallon .	×	·09983	= kilos. per litre.
Gallons per square foot	×	48·905	= litres per square metre.
Foot-pounds . . .	×	·1382	= kilogrammetres.
Foot-tons.	×	·3333	= tonne-metres.
Horse-power . . .	×	1·0139	= force de cheval.
Pounds per h.p. . .	×	·477	= kilos. per cheval.
Heat units	×	·252	= calories.

Metrical to English System.

Kilos. per metre	×	·672	= lbs. per foot.
Kilos. per metre	×	2·016	= lbs. per yard.
Kilos. per metre	×	·0003	= tons per foot.
Kilos. per metre	×	·0009	= tons per yard.
Kilos. per metre	×	3·548	= lbs. per mile.
Kilos. per square centimetre	×	14·223	= lbs. per square inch.
Kilos. per square millimetre	×	·635	= tons per square inch.
Kilos. per square metre .	×	·2048	= lbs. per square foot.
Tonnes per square metre .	×	·0914	= tons per square foot.
Tonnes per square metre .	×	·823	= tons per square yard.
Kilos. per cubic metre . .	×	1·686	= lbs. per cubic yard.
Kilos. per cubic metre . .	×	·0624	= lbs. per cubic foot.
Tonnes per cubic metre .	×	·752	= tons per cubic yard.
Grammes per litre . . .	×	73·09	= grains per gallon.
Kilos. per litre	×	10·438	= lbs. per gallon.
Kilogrammetres	×	7·233	= foot-pounds.
Tonne-metres	×	3·000	= foot-tons.
Force de cheval	×	·9863	= horse-power.
Kilos. per cheval . . .	×	2·235	= lbs. per horse-power.
Calories	×	3·968	= heat units.

THE FOLLOWING EQUIVALENTS OF METRIC WEIGHTS AND MEASURES IN TERMS OF IMPERIAL WEIGHTS AND MEASURES FOR USE IN TRADE WERE SANCTIONED BY AN ORDER IN COUNCIL ON THE 19TH MAY, 1898.

Metric to Imperial.

Linear Measure.

1 millimetre (mm.) ($\frac{1}{1000}$ m.)	=	0·03937 inch.
1 centimetre ($\frac{1}{100}$ m.) . .	=	0·3937 "
1 decimetre ($\frac{1}{10}$ m.) . .	=	3·937 inches.
1 metre (m.)	=	39·370113 inches. 3·280843 feet. 1·0936143 yards.
1 dekametre (10 m.) . .	=	10·936 yards.
1 hectometre (100 m.) .	=	109·36 "
1 kilometre (1000 m.) .	=	0·62137 mile.

Square Measure.

1 square centimetre	=	0·15500 square inch.
1 square decimetre (100 square centimetres)	=	15·500 square inches.
1 square metre (100 square decimetres) .	=	10·7639 square feet. 1·1960 square yards.
1 are (100 square metres)	=	119·60 square yards.
1 hectare (100 ares, or 10,000 square metres)	=	2·4711 acres.

Cubic Measure.

1 cubic centimetre	=	0·0610 cubic inch.
1 cubic decimetre (c.d.) (1000 cubic centimetres)	=	61·024 cubic inches.
1 cubic metre (1000 cubic decimetres) . . .	=	35·3148 cubic feet. 1·307954 cubic yards.

Measure of Capacity.

1 centilitre ($\frac{1}{100}$ litre)	=	0·070 gill.
1 decilitre ($\frac{1}{10}$ litre) .	=	0·176 pint.
1 litre	=	1·75980 pints.
1 dekalitre (10 litres)	=	2·200 gallons.
1 hectolitre (100 litres)	=	2·75 bushels.

Weight—Avoirdupois.

1 milligram ($\frac{1}{1000}$ grm.)	=	0·015 grain.
1 centigram ($\frac{1}{100}$ grm.)	=	0·154 "
1 decigram ($\frac{1}{10}$ grm.) .	=	1·543 grains.
1 gramme (1 grm.) .	=	15·432 "
1 dekagram (10 grm.)	=	5·644 drams.
1 hectogram (100 grm.)	=	3·527 oz.
1 kilogram (1000 grm.)	=	2·2046223 lb., or 15432·3564 grains.
1 myriagram (10 kilog.)	=	22·046 lb.
1 quintail (100 kilog.)	=	1·968 cwt.
1 tonne (1000 kilog.) .	=	·9842 ton.

Weight—Troy.

$$1 \text{ gram (1 grm.)} = \begin{cases} \cdot03215 \text{ oz. troy.} \\ 15\cdot432 \text{ grains.} \end{cases}$$

Weight—Apothecaries.

$$1 \text{ gram (1 grm.)} = \begin{cases} \cdot2572 \text{ drachm.} \\ \cdot7716 \text{ scruple.} \\ 15\cdot432 \text{ grains.} \end{cases}$$

EQUIVALENTS OF IMPERIAL AND METRIC WEIGHTS AND MEASURES.

Imperial to Metric.

Linear Measure.

1 inch	=	25·400 millimetres.
1 foot (12 inches) .	=	·30480 metre.
1 yard (3 feet) . .	=	·914399 metre.
1 fathom (6 feet) .	=	1·8288 metres.
1 pole (5½ yards) .	=	5·0292 "
1 chain (22 yards) .	=	20·1168 "
1 furlong (220 yards)	=	201·168 "
1 mile (8 furlongs)	=	1·6093 kilometres.

Square Measure.

1 square inch	=	6·4516 square centimetres.
1 square foot (144 square inches)	=	9·2903 square decimetres.
1 square yard (9 square feet) .	=	·836126 square metres.
1 perch (30¼ square yards) .	=	25·293 square metres.
1 rood (40 perches)	=	10·117 ares.
1 acre (4840 square yards) . .	=	·40468 hectare.
1 square mile (640 acres) . .	=	259·00 hectares.

Cubic Measure.

1 cubic inch	=	16·387 cubic centimetres.
1 cubic foot (1728 cubic inches)	=	·028317 cubic metre.
1 cubic yard (27 cubic feet) .	=	·764553 " "

Measures of Capacity.

1 gill	=	1·42 decilitres.
1 pint (4 gills) . .	=	·568 litre.
1 quart (2 pints) .	=	1·136 litres.
1 gallon (4 quarts)	=	4·5459631 litres.
1 peck (2 gallons) .	=	9·092 "
1 bushel (8 gallons)	=	3·637 dekalitres.
1 quarter (8 bushels)	=	2·909 hectolitres.

Apothecaries Measure.

1 minim	= ·059 millilitre.
1 fluid scruple	= 1·184 millilitres.
1 fluid drachm (60 minims) . . .	= 3·552 „
1 fluid ounce (8 drachms)	= 2·84123 centilitres.
1 pint	= ·568 litre.
1 gallon (8 pints or 160 fluid ounces)	= 4·5459631 litres.

Avoirdupois Weight.

1 grain	= ·0648 gramme.
1 dram	= 1·772 grams.
1 ounce (16 drams)	= 28·350 „
1 pound (16 oz. or 7000 grains)	= ·45359243 kilogram.
1 stone (14 lbs.)	= 6·350 kilograms.
1 quarter (28 lbs.)	= 12·70 „
1 hundredweight (cwt.) (112 lbs.)	= 50·80 „ ·5080 quintail.
1 ton (20 cwt.)	= 1·0160 tonnes or 1016 kilograms.

Troy Weight.

1 grain	= ·0648 gramme.
1 pennyweight (24 grains) .	= 1·5552 grammes.
1 troy ounce (20 pennyweights)	= 31·1035 „

Apothecaries Weight.

1 grain	= ·0648 gramme.
1 scruple (20 grains)	= 1·296 grammes.
1 drachm (3 scruples)	= 3·888 „
1 ounce (8 drachms)	= 31·1035 „

NOTE.—Approximately one litre equals 1000 cubic centimetres, and one millilitre equals 1·00016 cubic centimetres.

APPROXIMATE EQUIVALENTS.

1 millimetre	= $\frac{1}{25}$ inch.
1 metre . .	= 3 feet 3 inches and 3 eighths, or $1\frac{1}{11}$ yards.
1 kilometre .	= $\frac{5}{8}$ mile.
1 inch . . .	= $2\frac{1}{2}$ centimetres.
1 mile . . .	= $1\frac{3}{5}$ kilometre.

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1 square inch	= $6\frac{1}{2}$ square centimetres.
1 square metre	= $1\frac{1}{5}$ square yard, or $10\frac{3}{4}$ square feet.
1 square yard	= $\frac{5}{6}$ square metre.
1 acre . . .	= 4000 square metres.

1 cubic yard	$= \frac{3}{4}$ cubic metre.
1 cubic metre	$= 1\frac{1}{3}$ cubic yard.
1 litre . . .	$= 1\frac{3}{4}$ pints.
1 gallon . .	$= 4\frac{1}{2}$ litres.
1 cubic foot .	$= 28{\cdot}3$ litres.

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1 gramme .	$= 15\frac{1}{2}$ grains.
1 kilogram .	$= 2\frac{1}{5}$ pounds.
1000 kilograms	$=$ 1 English ton.
1 cwt. . . .	$=$ 51 kilograms

TABLE 46.

SINES OF ANGLES OF AN EQUALLY DIVIDED CIRCLE WHOSE RADIUS IS 1.

N	$\frac{360^\circ}{N}$ °	′	″	$\frac{180^\circ}{N}$ °	′	″	Sin $\frac{180^\circ}{N}$	N	$\frac{360^\circ}{N}$ °	′	″	$\frac{180^\circ}{N}$ °	′	″	Sin $\frac{180^\circ}{N}$
1	360	0	0	180	0	0	1·0	51	7	3	32	3	31	46	·06156
2	180	0	0	90	0	0	1·0	52	6	55	23	3	27	42	·06038
3	120	0	0	60	0	0	·86603	53	6	47	33	3	23	46	·05922
4	90	0	0	45	0	0	·70711	54	6	40	0	3	20	0	·05814
5	72	0	0	36	0	0	·58799	55	6	32	44	3	16	22	·05709
6	60	0	0	30	0	0	·50000	56	6	25	43	3	12	51	·05607
7	51	25	43	25	42	51	·43388	57	6	18	57	3	9	28	·05509
8	45	0	0	22	30	0	·38268	58	6	12	25	3	6	12	·05414
9	40	0	0	20	0	0	·34202	59	6	6	6	3	3	3	·05322
10	36	0	0	18	0	0	·30902	60	6	0	0	3	0	0	·05234
11	32	43	38	16	21	49	·28173	61	5	54	6	2	57	3	·05147
12	30	0	0	15	0	0	·25882	62	5	48	23	2	54	12	·05065
13	27	41	32	13	50	46	·23931	63	5	42	51	2	51	26	·04985
14	25	42	51	12	51	26	·22252	64	5	37	30	2	48	45	·04907
15	24	0	0	12	0	0	·20791	65	5	32	18	2	46	9	·04831
16	22	30	0	11	15	0	·19509	66	5	27	17	2	43	38	·04758
17	21	10	35	10	35	18	·18375	67	5	22	23	2	41	12	·04688
18	20	0	0	10	0	0	·17365	68	5	17	39	2	38	49	·04618
19	18	56	50	9	28	25	·16454	69	5	13	3	2	36	31	·04551
20	18	0	0	9	0	0	·15643	70	5	8	34	2	34	17	·04486
21	17	8	34	8	34	17	·14904	71	5	4	14	2	32	7	·04423
22	16	21	49	8	10	55	·14233	72	5	0	0	2	30	0	·04362
23	15	39	8	7	49	34	·13616	73	4	55	53	2	27	57	·04303
24	15	0	0	7	30	0	·13053	74	4	51	53	2	25	57	·04245
25	14	24	0	7	12	0	·12533	75	4	48	0	2	24	0	·04188
26	13	50	46	6	55	23	·12055	76	4	44	13	2	22	6	·04132
27	13	20	0	6	40	0	·11609	77	4	40	31	2	20	16	·04079
28	12	51	26	6	25	43	·11197	78	4	36	55	2	18	27	·04026
29	12	24	50	6	12	25	·10812	79	4	33	25	2	16	43	·03976
30	12	0	0	6	0	0	·10453	80	4	30	0	2	15	0	·03926
31	11	36	46	5	48	23	·10117	81	4	26	40	2	13	20	·03878
32	11	15	0	5	37	30	·09801	82	4	23	25	2	11	42	·03830
33	10	54	33	5	27	16	·09506	83	4	20	14	2	10	7	·03784
34	10	35	18	5	17	39	·09227	84	4	17	9	2	8	34	·03739
35	10	17	8	5	8	34	·08963	85	4	14	7	2	7	4	·03695
36	10	0	0	5	0	0	·08716	86	4	11	10	2	5	35	·03652
37	9	43	47	4	51	54	·08510	87	4	8	17	2	4	8	·03610
38	9	28	25	4	44	13	·08258	88	4	5	27	2	2	44	·03569
39	9	13	51	4	36	55	·08047	89	4	2	42	2	1	21	·03529
40	9	0	0	4	30	0	·07846	90	4	0	0	2	0	0	·03490
41	8	46	50	4	23	25	·07655	91	3	57	22	1	58	41	·03446
42	8	34	17	4	17	9	·07473	92	3	54	46	1	57	23	·03414
43	8	22	20	4	11	10	·07300	93	3	52	15	1	56	8	·03378
44	8	10	55	4	5	27	·07134	94	3	49	47	1	54	54	·03342
45	8	0	0	4	0	0	·06976	95	3	47	22	1	53	41	·03307
46	7	49	34	3	54	47	·06825	96	3	45	0	1	52	30	·03272
47	7	39	34	3	49	47	·06679	97	3	42	41	1	51	20	·03238
48	7	30	0	3	45	0	·06540	98	3	40	24	1	50	12	·03205
49	7	20	49	3	40	24	·06407	99	3	38	11	1	49	5	·03172
50	7	12	0	3	36	0	·06279	100	3	36	0	1	48	0	·03141

TABLE 47.

Speeds per Hour in Miles and Kilometres.

Time of one mile. Min.	Sec.	Miles per hour.	Kiloms. per hour.	Time of one mile. Min.	Sec.	Miles per hour.	Kiloms. per hour.	Time of one mile. Min.	Sec.	Miles per hour.	Kiloms. per hour.	Time of one mile. Min.	Sec.	Miles per hour.	Kiloms. per hour.
1	0	60	96·5	1	32	39·1	63	2	12	27·3	43·9	4	36	13	20·9
1	1	59	94·9	1	33	38·7	62·3	2	15	26·7	42·9	4	42	12·8	20·6
1	2	58	93·3	1	34	38·3	61·6	2	18	26·1	42	4	48	12·5	20·1
1	3	57·1	91·7	1	35	37·9	61	2	21	25·5	41·1	4	54	12·2	19·6
1	4	56·3	90·5	1	36	37·5	60·4	2	24	25	40·2	5	0	12	19·3
1	5	55·4	89·1	1	37	37·1	59·7	2	27	24·5	39·4	5	12	11·5	18·5
1	6	54·5	87·6	1	38	36·7	59	2	30	24	38·6	5	24	11·1	17·8
1	7	53·7	86·4	1	39	36·4	58·5	2	33	23·6	37·9	5	36	10·7	17·2
1	8	53	85·2	1	40	36	57·9	2	36	23·1	37·1	5	48	10·3	16·6
1	9	52·2	84	1	41	35·7	57·4	2	39	22·6	36·3	6	0	10	16·1
1	10	51·4	82·7	1	42	35·3	56·8	2	42	22·2	35·7	6	15	9·6	15·4
1	11	50·7	81·5	1	43	34·9	56·2	2	45	21·8	35·1	6	30	9·2	14·8
1	12	50	80·4	1	44	34·6	55·7	2	48	21·4	34·4	6	45	8·9	14·3
1	13	49·4	79·4	1	45	34·3	55·2	2	51	20·1	33·9	7	0	8·6	13·8
1	14	48·6	78·3	1	46	34	54·7	2	54	20·7	33·3	7	15	8·3	13·4
1	15	48	77·2	1	47	33·7	54·2	2	57	20·3	32·7	7	30	8	12·9
1	16	47·4	76·1	1	48	33·4	53·7	3	0	20	32·2	7	45	7·7	12·4
1	17	46·7	75·1	1	49	33	53·1	3	6	19·4	31·2	8	0	7·5	12
1	18	46·2	74·2	1	50	32·7	52·6	3	12	18·8	30·2	8	30	7·1	11·4
1	19	45·6	73·3	1	51	32·4	52·1	3	18	18·2	29·3	9	0	6·7	10·8
1	20	45	72·4	1	52	32·1	51·6	3	24	17·7	28·5	9	30	6·3	10·1
1	21	44·4	71·5	1	53	31·8	51·1	3	30	17·1	27·3	10	0	6	9·6
1	22	43·9	70·6	1	54	31·6	50·7	3	36	16·7	26·9	10	30	5·7	9·2
1	23	43·3	69·7	1	55	31·3	50·3	3	42	16·2	26	11	0	5·5	8·8
1	24	42·8	68·9	1	56	31	49·9	3	48	15·7	25·2	11	30	5·2	8·4
1	25	42·4	68·1	1	57	30·8	49·6	3	54	15·4	24·8	12	0	5	8
1	26	41·9	67·4	1	58	30·5	49·1	4	0	15	24·2	13	0	4·6	7·4
1	27	41·4	66·6	1	59	30·2	48·7	4	6	14·6	23·3	14	0	4·3	6·9
1	28	40·9	65·8	2	0	30	48·3	4	12	14·3	23	15	0	4	6·4
1	29	40·4	65	2	3	29·2	47	4	18	13·9	22·4	20	0	3	4·8
1	30	40	64·4	2	6	28·6	46	4	24	13·6	21·9	30	0	2	3·2
1	31	39·6	63·7	2	9	27·9	44·9	4	30	13·3	21·4	60	0	1	1·6

TABLE 48.

Weight of Seamless Copper Tubes (Imperial Wire Gauge, 1884).

The Broughton Copper Company, Limited, Manchester.

1884. I.W.G.		Thickness of copper.																			
		0000	000	00	0	1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
Inches		·400	·372	·348	·324	·300	·276	·252	·232	·212	·192	·176	·160	·144	·128	·116	·104	·092	·080	·072	·064
		$\frac{13}{32}$ B	$\frac{3}{8}$ B	$\frac{11}{32}$ F	$\frac{21}{64}$ B	$\frac{19}{64}$ F	$\frac{9}{32}$ B	$\frac{1}{4}$ F	$\frac{15}{64}$ B	$\frac{7}{32}$ B	$\frac{3}{16}$ F	$\frac{11}{64}$ F	$\frac{5}{32}$ F	$\frac{9}{64}$ F	$\frac{1}{8}$ F	$\frac{7}{64}$ F	$\frac{7}{64}$ B	$\frac{3}{32}$	$\frac{5}{64}$ F	$\frac{5}{64}$ B	$\frac{1}{16}$ F
Millimrs.		10·160	9·449	8·839	8·229	7·620	7·010	6·401	5·893	5·385	4·877	4·470	4·064	3·658	3·251	2·946	2·642	2·337	2·032	1·829	1·626
Inter. diam.		Weight of a lineal foot in pounds.																			
Ins.	Mills.																				
1	25·4	6·77	6·17	5·67	5·19	4·72	4·26	3·82	3·46	3·11	2·77	2·50	2·24	1·99	1·75	1·57	1·39	1·21	1·04	0·93	0·82
1⅛	28·6	7·38	6·74	6·20	5·68	5·17	4·68	4·20	3·81	3·43	3·06	2·77	2·49	2·21	1·94	1·74	1·55	1·35	1·17	1·04	0·92
1¼	31·7	7·98	7·30	6·73	6·17	5·62	5·09	4·58	4·16	3·75	3·35	3·04	2·73	2·43	2·13	1·92	1·70	1·49	1·29	1·15	1·02
1⅜	34·9	8·59	7·86	7·25	6·66	6·08	5·51	4·96	4·51	4·07	3·64	3·30	2·97	2·65	2·33	2·09	1·86	1·63	1·41	1·26	1·11
1½	38·1	9·19	8·42	7·78	7·15	6·53	5·93	5·34	4·86	4·39	3·93	3·57	3·21	2·86	2·52	2·27	2·02	1·77	1·53	1·37	1·21
1⅝	41·3	9·80	8·99	8·31	7·64	6·99	6·35	5·72	5·21	4·71	4·22	3·83	3·45	3·08	2·71	2·44	2·17	1·91	1·65	1·48	1·31
1¾	44·4	10·40	9·55	8·83	8·13	7·44	6·76	6·10	5·56	5·03	4·51	4·10	3·70	3·30	2·91	2·62	2·33	2·05	1·77	1·59	1·40
1⅞	47·6	11·01	10·11	9·36	8·62	7·89	7·18	6·48	5·91	5·35	4·80	4·37	3·94	3·52	3·10	2·79	2·49	2·19	1·89	1·70	1·50
2	50·8	11·61	10·67	9·88	9·11	8·35	7·60	6·86	6·26	5·67	5·09	4·63	4·18	3·73	3·29	2·97	2·65	2·33	2·01	1·80	1·60
2⅛	54·0	12·22	11·24	10·41	9·60	8·80	8·02	7·25	6·61	5·99	5·38	4·90	4·42	3·95	3·49	3·14	2·80	2·47	2·13	1·91	1·69
2¼	57·1	12·82	11·80	10·94	10·09	9·25	8·43	7·63	6·97	6·31	5·67	5·16	4·66	4·17	3·68	3·32	2·96	2·61	2·25	2·02	1·79
2⅜	60·3	13·43	12·36	11·46	10·58	9·71	8·85	8·01	7·32	6·63	5·96	5·43	4·91	4·39	3·88	3·50	3·12	2·75	2·38	2·13	1·89
2½	63·5	14·03	12·92	11·99	11·07	10·16	9·27	8·39	7·67	6·95	6·25	5·70	5·15	4·61	4·07	3·67	3·28	2·88	2·50	2·24	1·98
2⅝	66·7	14·64	13·49	12·52	11·56	10·62	9·69	8·77	8·02	7·28	6·54	5·96	5·39	4·82	4·26	3·85	3·43	3·02	2·62	2·35	2·08
2¾	69·8	15·24	14·05	13·04	12·05	11·07	10·10	9·15	8·37	7·60	6·83	6·23	5·63	5·04	4·46	4·02	3·59	3·16	2·74	2·46	2·18
2⅞	73·0	15·85	14·61	13·57	12·54	11·52	10·52	9·53	8·72	7·92	7·12	6·50	5·87	5·26	4·65	4·20	3·75	3·30	2·86	2·57	2·27

3	76·2	16·45	15·17	14·09	13·03	11·98	10·94	9·91	9·07	8·24	7·41	6·76	6·12	5·48	4·84	4·37	3·90	3·44	2·98	2·68	2·37
3¼	82·5	17·66	16·30	15·15	14·01	12·88	11·77	10·68	9·77	8·88	7·99	7·29	6·60	5·91	5·23	4·72	4·22	3·72	3·22	2·89	2·57
3½	88·9	18·87	17·42	16·20	14·99	13·79	12·61	11·44	10·47	9·52	8·58	7·83	7·08	6·35	5·62	5·07	4·53	4·00	3·46	3·11	2·76
3¾	95·2	20·08	18·55	17·25	15·97	14·70	13·44	12·20	11·18	10·16	9·16	8·36	7·57	6·78	6·00	5·42	4·85	4·28	3·71	3·33	2·95
4	101·6	21·29	19·67	18·30	16·95	15·61	14·28	12·96	11·88	10·80	9·74	8·89	8·05	7·22	6·39	5·78	5·16	4·55	3·95	3·55	3·15
4¼	107·9	22·50	20·80	19·36	17·93	16·51	15·11	13·72	12·58	11·44	10·32	9·42	8·54	7·65	6·78	6·13	5·48	4·83	4·19	3·76	3·34
4½	114·3	23·71	21·93	20·41	18·91	17·42	15·95	14·49	13·28	12·08	10·90	9·96	9·02	8·09	7·17	6·48	5·79	5·11	4·43	3·98	3·53
4¾	120·6	24·92	23·05	21·46	19·89	18·33	16·78	15·25	13·98	12·73	11·48	10·49	9·50	8·53	7·55	6·83	6·11	5·39	4·67	4·20	3·73
5	127·0	26·13	24·18	22·51	20·87	19·23	17·62	16·01	14·68	13·37	12·06	11·02	9·99	8·96	7·94	7·18	6·42	5·67	4·92	4·42	3·92
5¼	133·3	27·34	25·30	23·57	21·85	20·14	18·45	16·77	15·39	14·01	12·64	11·55	10·47	9·40	8·33	7·53	6·74	5·95	5·16	4·64	4·11
5½	139·7	28·55	26·43	24·62	22·83	21·05	19·29	17·54	16·09	14·65	13·22	12·08	10·96	9·83	8·71	7·88	7·05	6·22	5·40	4·85	4·31
5¾	146·0	29·76	27·55	25·67	23·81	21·96	20·12	18·30	16·79	15·29	13·80	12·62	11·44	10·27	9·10	8·23	7·36	6·50	5·64	5·07	4·50
6	152·4	30·97	28·68	26·72	24·79	22·86	20·95	19·06	17·49	15·93	14·38	13·15	11·92	10·70	9·49	8·58	7·68	6·78	5·88	5·29	4·69
6¼	158·7	32·18	29·80	27·78	25·77	23·77	21·79	19·82	18·19	16·57	14·96	13·68	12·41	11·14	9·88	8·93	7·99	7·06	6·13	5·51	4·89
6½	165·1	33·39	30·93	28·83	26·75	24·68	22·62	20·58	18·89	17·21	15·54	14·21	12·89	11·57	10·26	9·28	8·31	7·34	6·37	5·72	5·08
6¾	171·4	34·60	32·05	29·88	27·73	25·59	23·46	21·35	19·60	17·85	16·12	14·75	13·37	12·01	10·65	9·63	8·62	7·61	6·61	5·94	5·28
7	177·8	35·81	33·18	30·93	28·71	26·49	24·29	22·11	20·30	18·50	16·70	15·28	13·86	12·44	11·04	9·99	8·94	7·89	6·85	6·16	5·47
7¼	184·1	37·02	34·30	31·99	29·69	27·40	25·13	22·87	21·00	19·14	17·29	15·81	14·34	12·88	11·42	10·34	9·25	8·17	7·09	6·38	5·66
7½	190·5	38·23	35·43	33·04	30·67	28·31	25·96	23·63	21·70	19·78	17·87	16·34	14·83	13·32	11·81	10·69	9·57	8·45	7·34	6·60	5·86
7¾	196·8	39·44	36·55	34·09	31·65	29·22	26·80	24·39	22·40	20·42	18·45	16·88	15·31	13·75	12·20	11·04	9·88	8·73	7·58	6·81	6·05
8	203·2	40·65	37·68	35·14	32·63	30·12	27·63	25·16	23·10	21·06	19·03	17·41	15·79	14·19	12·59	11·39	10·20	9·01	7·82	7·03	6·24
8¼	209·5	41·86	38·80	36·20	33·61	31·03	28·47	25·92	23·81	21·70	19·61	17·94	16·28	14·62	12·97	11·74	10·51	9·28	8·06	7·25	6·44
8½	215·9	43·07	39·93	37·25	34·59	31·94	29·30	26·68	24·51	22·34	20·19	18·47	16·76	15·06	13·36	12·09	10·82	9·56	8·30	7·47	6·63
8¾	222·2	44·28	41·05	38·30	35·57	32·84	30·14	27·44	25·21	22·98	20·77	19·00	17·25	15·49	13·75	12·44	11·14	9·84	8·55	7·68	6·82
9	228·6	45·49	42·18	39·35	36·55	33·75	30·97	28·21	25·91	23·63	21·35	19·54	17·73	15·93	14·13	12·79	11·45	10·12	8·79	7·90	7·02
9¼	234·9	46·70	43·30	40·41	37·53	34·66	31·81	28·97	26·61	24·27	21·93	20·07	18·21	16·36	14·52	13·14	11·77	10·40	9·03	8·12	—
9½	241·3	47·91	44·43	41·46	38·51	35·57	32·64	29·73	27·31	24·91	22·51	20·60	18·70	16·80	14·91	13·49	12·08	10·68	9·27	8·34	—
9¾	247·6	49·12	45·55	42·51	39·49	36·47	33·48	30·49	28·02	25·55	23·09	21·13	19·18	17·24	15·30	13·84	12·40	10·95	9·51	8·55	—
10	254·0	50·33	46·68	43·56	40·47	37·38	34·31	31·25	28·72	26·19	23·67	21·67	19·67	17·67	15·68	14·20	12·71	11·23	9·76	8·77	—

F = full. B = bare.

INDEX.

A.

B.

C.

G.

H.

I.

V.

W.

INDEX TO FORMULÆ.

THE END.

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